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LIQUID HYDROGEN RECOVERY SYSTEM STUDY

By

K. L. Bush, G. V. Schwent, and H. G. Starck

Prepared For

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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Technical Management
NASA Lewis Research Center
Cleveland, Ohio
M-1 Turbo-Machinery Section
Ward W. Wilcox

0122762 AIRESEARCH Mfg. Co., MANUFACTURING COMPANY
LOS ANGELES, CALIFORNIA
PHOENIX, ARIZONA

LIQUID HYDROGEN RECOVERY SYSTEMS STUDY

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ABSTRACT

A

The M-1 pump test facility at Aerojet, Sacramento, California can be significantly improved by the addition of either a heat exchanger or a turbine power absorber to the present recovery system. The heat exchanger system will cost about $900, 000 and can save up to $9, 000, 000 in liquid hydrogen over a 5-year period. A turbine power absorber unit will cost about $1,700,000 and has a savings potential of $20,000,000 for the same period. The timing for fabrication and installation are 15 and 24 months, respectively.

~~AUTHOR~~

LIQUID HYDROGEN RECOVERY SYSTEM STUDY

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K. L. Bush, G. V. Schwent, and H. G. Starck

AiResearch Manufacturing Company

SUMMARY

The primary objective of this contract is to perform a study of the technical and economic aspects of various liquid hydrogen recovery systems in sufficient depth to determine the relative merits and disadvantages of the various systems. The study was based on the planned operations for Test Stand E-2 at the Aerojet-General Corporation, Liquid Rocket Plant, Sacramento, California.

The study indicates that two different type systems, a 60,000 hp turbine-power absorber system, and a 30-ton heat exchanger (using boiloff vent gases for cooling) are technically and economically feasible for improving the recovery characteristics of the Aerojet M-1 pump test stand.

The turbine-power absorber system will cost about \$1,7~~5~~00,000 and can save about \$20,000,000 at a cost of \$0.50 per gallon for LH_2. The heat exchanger system will cost about \$900,000 and can save about \$9,000,000. Even at double the costs of the systems and at half-price of the LH_2, the potential net savings are clearly substantial.

The timing, which includes installation and procurement of materials is estimated to be about 15 months for the heat exchanger and 24 months for the turbine and compressor.

The recommended turbine system consists of a dual flow, 7-stage axial turbine located at the upstream orifice of the M-1 pump discharge line. The power generated by the turbine is absorbed by two, double suction, centrifugal air compressors. The basic controls will consist of a 12-in. pressure control valve in a bypass line around the turbine and a 12-in. bypass control valve at the downstream orifice near the catch tank. Each air compressor will have a 14-in. modulating valve at the exit. The above costs include two turbine units and three compressor units, plus spares for certain critical parts such as bearings, seals, and valve actuators. Since there is no way of testing

the turbine except in its installed position at Aerojet, there is no associated development program, and unit costs are correspondingly low. Therefore, a very conservative design approach must be taken to assure a satisfactory performance. With the provision of two machines plus critical spare parts, a satisfactory level of reliability is anticipated for this installation.

The heat exchanger unit will consist of a finned tube, folded core design, made of aluminum and suspended in an aluminum vacuum-jacketed tank. It should be located near the top of the E-2 catch tank on a separate support and will require an extensive modification to the present vent system. The only control is a 12-in. bypass valve at the present intake orifice at the catch tank. Again, there is no way of testing the total performance of this unit prior to its installation. However, this is not a problem area since an overdesign can only result in an improved performance, particularly if the liquid entrainment in the vent gas is a large percentage of the flow rate. Therefore, a high reliability and performance can be assured by an overdesign approach for this unit.

INTRODUCTION

This report presents the results of a study of liquid hydrogen recovery systems as required by study Contract NAS 3-2751.

The requirements to be fulfilled by the study are given in terms of six tasks in Contract NAS 3-2751. Task I consists of the selection of the most promising type of recovery system from various combinations of turbines, power absorbers, and heat exchangers. Task II requires a full preliminary design of the selected system, including controls. Tasks III and V are to consider the feedback effect of the recovery system operation on the E-2 test stand and the procedures required for satisfactory operation, and Tasks IV and VI are concerned with the economic aspects and possible schedule for integration with the E-2 test program. A single design point is given for the output of Test Stand E-2 as a basis for the design study and consists of the following items:

1. Frequency of tests: 7 runs per month for 3 to 5 years

2. Duration of each test: 300 seconds

3. Hydrogen pump outlet pressure: 1800 psi

4. Hydrogen pump outlet temperature: 50° to 56°R

5. Flow rate: 60, 000 gpm

The contract further requires that additional information as may be made available by the LVPO technical manager, shall be utilized to the fullest extent practicable.

In order to attain the objectives and fulfill the requirements of the above statement of work, a well-defined approach has been taken. This approach, or guide line, may be summarized by the following statements:

1. Obtain detailed information on the E-2 test stand and the pump test program so that installation problems and the effect of off-design operating conditions can be analyzed.

2. Determine the primary and secondary parameters which affect system performance and design.

3. Design the optimum system for achieving the objectives given in the contract statement of work.

4. Estimate component installation and maintenance costs and compare with potential savings of liquid hydrogen.

5. Determine the development risks and the effect of added equipment on recovery system safety and reliability.

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The report is divided into nine sections and five appendicies, which are as follows:

<u>Section</u>

I Recovery System Thermodynamics

II Aerojet E-2 Area Test Facility

III Turbine Design Studies

I RECOVERY SYSTEM THERMODYNAMICS

1. Simple (Throttling) Recovery System

When a high pressure fluid has an energy content which is less than that of a low pressure saturated vapor, the fluid may be dumped into a (low pressure) tank where a portion of the fluid may be "recovered" as a low pressure liquid. Figure 1A shows a diagram of this system, which is known as a "throttling process" since the high pressure fluid is generally ducted to the tank at a pressure above the vapor pressure, and then throttled by a valve or orifice. At the reduced pressure, the fluid "flashes" and a separation of liquid and vapor takes place. The high pressure energy is thus absorbed by the vapor which is released from the tank by a vent line.

Another method of obtaining the same result is to use a heat exchanger as a "subcooler" (Figure 1B) in which a portion of the tank liquid is vaporized to cool the incoming fluid so that the total delivery is in the liquid state. If the vapor is not superheated, the required vapor loss is the same in either case, the heat exchanger serving only to reduce the turbulence of the incoming fluid into the tank.

In the throttling or subcooler systems, the specific recovery can be expressed in terms of the specific enthalpies of the high and low pressure fluids

$$m_o = 1 - \frac{h - h_f}{h_{f_g}} \qquad (1)$$

where h = enthalpy of the high energy fluid

h_f = enthalpy of the tank liquid

h_{f_g} = enthalpy of vaporization

m_o = ratio of weight of liquid to weight of fluid delivered into the tank

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Section (continued)

Appendix

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Addendum

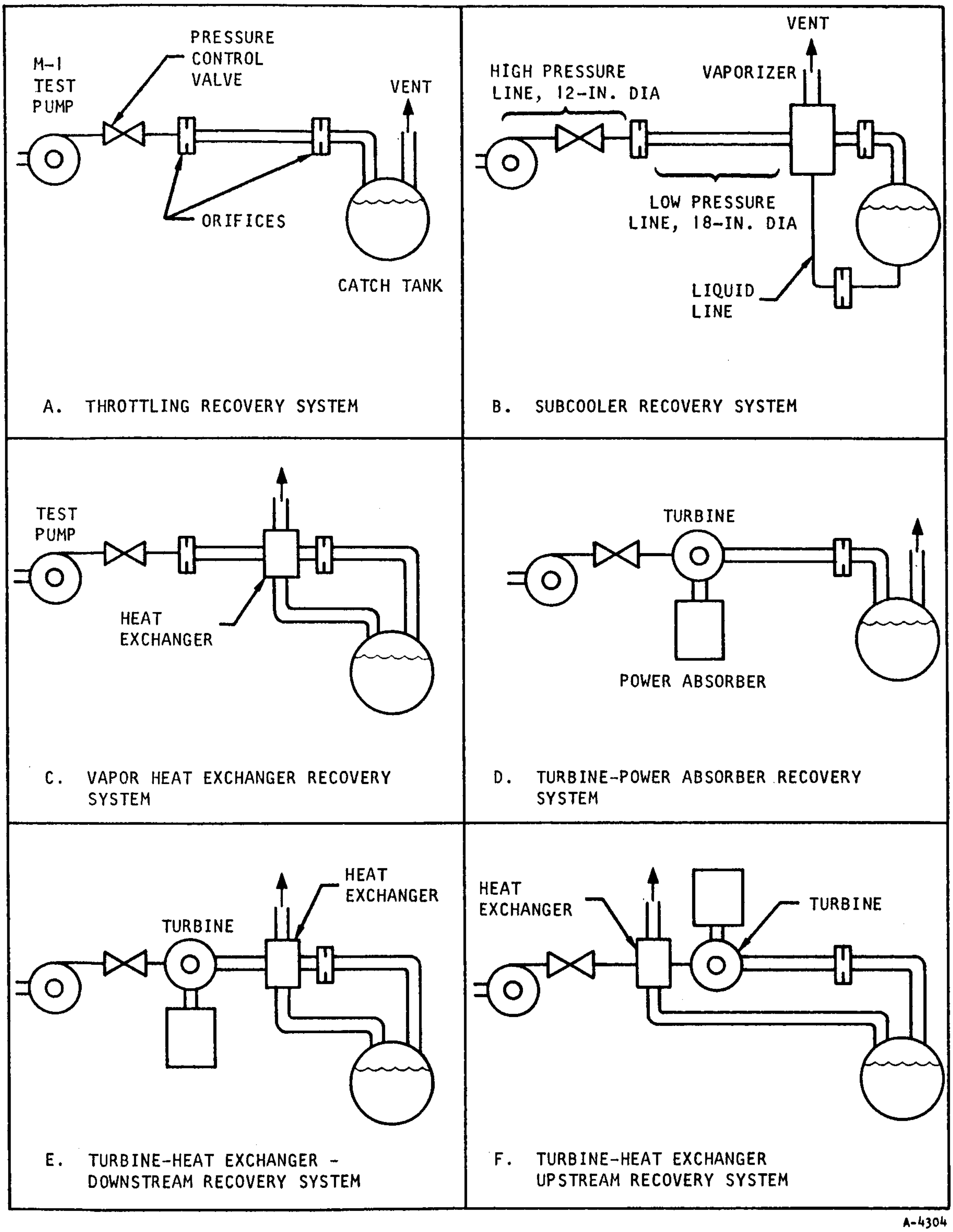

Figure 1. Schematic Diagrams of Recovery Systems

Since m_o cannot be negative, the limiting energy content of the fluid is $h < h_f + h_{fg}$.

These relations are most readily evaluated by means of the Mollier charts of fluid properties.

A pH diagram of the logic processes is shown in Figure 2. The initial condition of the fluid into the pump is h_o and the outlet state is

$$h - h_o = \frac{\Delta P}{\rho_{av} J \eta_P}$$

where ΔP = pressure rise

ρ_{av} = average density

η_p = pump efficiency

In the transfer line from the pump to the tank, the process is primarily one of constant enthalpy due to the relatively low velocities and well-insulated lines. The fluid arrives at the tank at nearly the same energy content as at the pump discharge, as represented by the vertical line from h to m_o. Within the tank the total energy content is unchanged, however, the phase separation results in the fluid having an energy content, h_f, and the vapor having an energy content, h_g, under the equilibrium conditions which will exist after the shutdown of the pump.

Tank Pressure

During operation, the pressure of the recovery tank affects the amount of liquid recovery due to the variation of fluid properties within the two-phase region. In general, a higher pressure results in less vapor formation and a greater recovery of liquid, except near the saturated vapor line.

For the simple throttling system, this variation is of no consequence since the amount of liquid which is ultimately available depends only upon the final "use" pressure of the tank. A greater recovery during operating transients simply boils off as the pressure falls to the

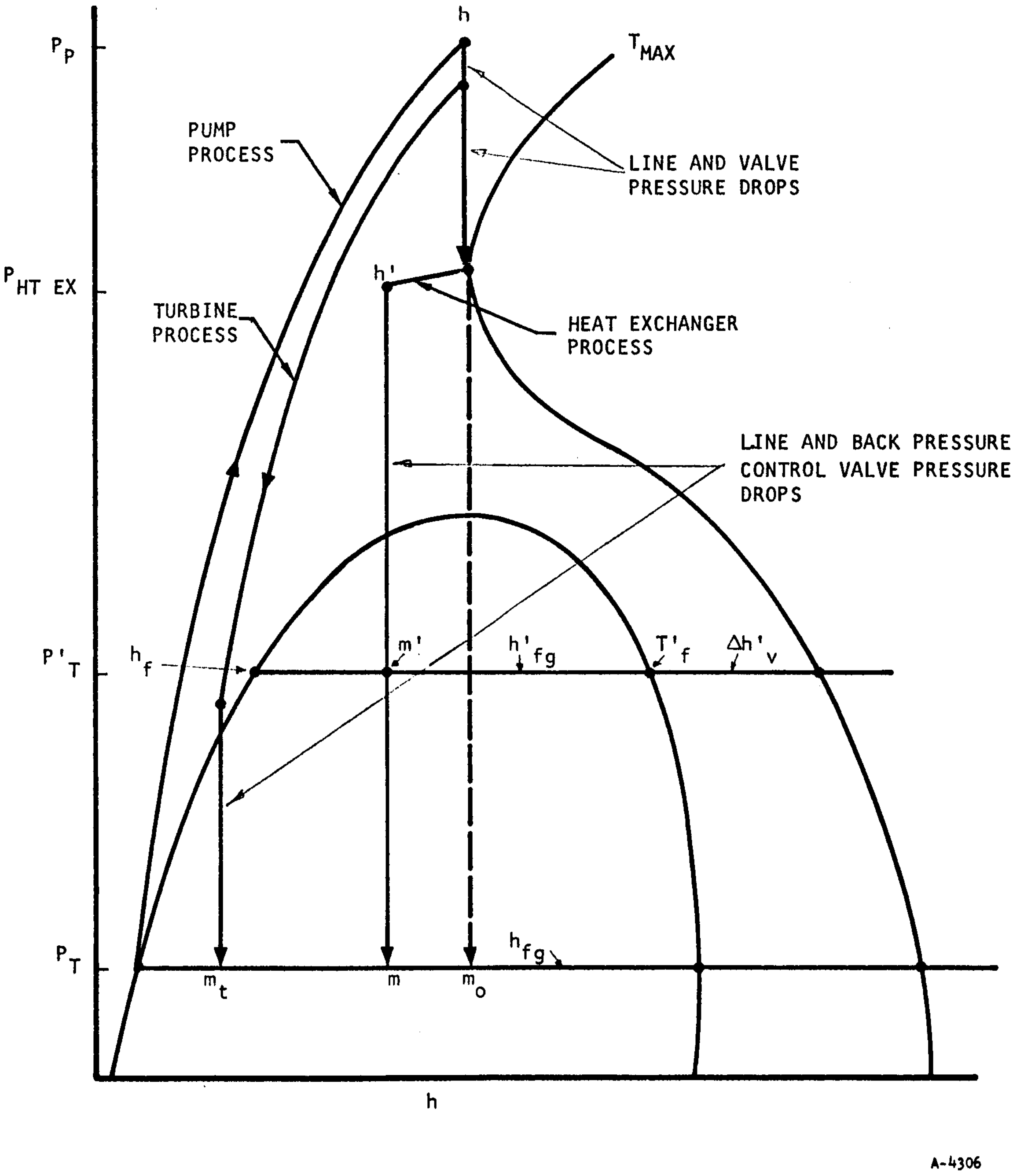

Figure 2. Pressure-Enthalpy Diagram of Recovery System Processes

final (usually ambient) pressure after the flow into the tank has stopped. Therefore, the basic parameters for the recovery of liquid in a simple system are the fluid's initial energy content h, and the final tank pressure which determines the value of h_f and h_{fg}.

Liquid Entrainment

Liquid entrainment in the vent gas represents an additional loss of liquid for the throttling system and is a function of the droplet size and turbulent velocities within the tank. The turbulent velocities will vary with tank size, flow rates, and liquid level, and the resulting entrainment is generally an indeterminant quantity which must be obtained from tests. Baffles and centrifuge-type flows are employed to minimize this loss.

The subcooler will reduce entrainment losses substantially if the heat exchanger has a high effectiveness and the catch tank pressure is equal to the saturation pressure of the incoming fluid.

2. Vapor Heat Exchanger Recovery System

The boiloff vapor in the tank is considerably cooler than the incoming fluid and this temperature difference can be utilized in a heat exchanger to improve the recovery rate of the system. The heat exchanger must be located near the tank, as shown in Figure 1C to obtain a maximum temperature difference and cooling of the incoming fluid. The cooling results in an increase in liquid recovery, and the amount of improvement depends upon the temperature difference between the tank vapor and the high energy fluid, the amount of boiloff vapor and the effectiveness of the heat exchanger. The temperature difference is a function of the pressures in the tank and in the heat exchanger during operation. The basic thermodynamic process is shown in Figure 2. If the tank operating pressure is higher than the final "use" pressure further boiloff reduces the recovery, as discussed in detail in Appendix A. This final recovery is expressed:

$$m = 1 - \frac{h - h_f - \dfrac{h - h_f'}{1 + h_{fg}'/E_v \; h_v'}}{h_{fg}} \qquad (2)$$

where primes indicate enthalpies at tank operating pressures

E_v = heat exchanger vapor side effectiveness

$\Delta h_v' \quad = \quad C_p \, (T_{max} - T_f')$

C_p = specific heat of tank vapor

T_{max} = temperature of high pressure fluid into the heat exchanger

T_f' = tank vapor temperature at operating pressure

By combining Equations (1) and (2), the additional recovery provided by the heat exchanger is obtained in terms of operating and final conditions:

$$m - m_o = \frac{h - h_f'}{h_{fg}(1 + h_{fg}'/E_v \Delta h_v')}$$

It should be noted that the heat exchanger will improve recovery at energy levels beyond those at which $m_o = 0$. Here the upper limit is determined by the differences in the specific heats of the vapor and the high pressure fluid, and the effectiveness of the heat exchanger. These relations may be expressed by

$$h_{max} \, f(T) \leq h_{fg} + E \, \bar{C}_p (T - T_o)$$

where $h_{max} \, f(T)$ = maximum energy level of incoming fluid as determined by its temperature, T

$\bar{C}_p$ = average specific heat of vapor between temperature T and T_o

E_v = heat exchanger effectiveness

T = temperature of high energy fluid

T_o = temperature of vapor

a. <u>Tank Pressure Effect</u>

In general, the higher the tank pressure, the lower the amount of recovery. This is due to the adverse effect of pressure on the heat capacity, $\Delta h_V'$, of the vapor and the subsequent boiling of the recovered liquid as the pressure falls to the final tank pressure after shutdown. As shown in Figure 3, the heat capacity of the vapor reduces with increased pressure, particularly at low temperatures. Thus, a substantial loss in vapor may occur after shutdown. For a decay from 60 psi to ambient, the boiloff, or loss in recovery, is 10 to 15 percent, which is equivalent to most of the potential gain of the heat exchanger.

From these considerations, it is evident that the tank pressure should be as near the final tank pressure as possible. This means that the vapor side pressure drops in the heat exchanger and vent line are critical. A large vent line diameter (compared to that of the simple recovery system) and a large heat transfer surface area are required for maximizing the recovery.

b. <u>Heat Exchanger Pressure Effect</u>

The temperature difference is greatest when the high pressure fluid is in the region of the Joule-Thomson inversion, which occurs near the dome in the supercritical region of the fluid. For liquid hydrogen, this region occurs at 200 to 1200 psi pressures, a good average being about 500 psi, as shown in Figure 4. Thus, the inlet pressure to the heat exchanger should be regulated to this range for maximum recovery.

c. <u>Heat Exchanger Performance</u>

Generally, a high effectiveness is warranted for this type of system. Then, the variation of E_V with the vapor flow rate, $(1 - m)$, will be relatively small and the use of an average value of E_V for the range of fluid conditions will yield information which is adequate for preliminary design or system evaluation purposes. Figure 5 shows the performance of a heat exchanger system as compared to a throttling system. The calculation was based upon the maximum temperature differences and an effectiveness of 0.95. It is evident that this system offers a substantial improvement in recovery, particularly at the high pressure and temperature conditions. At the test pump design point,

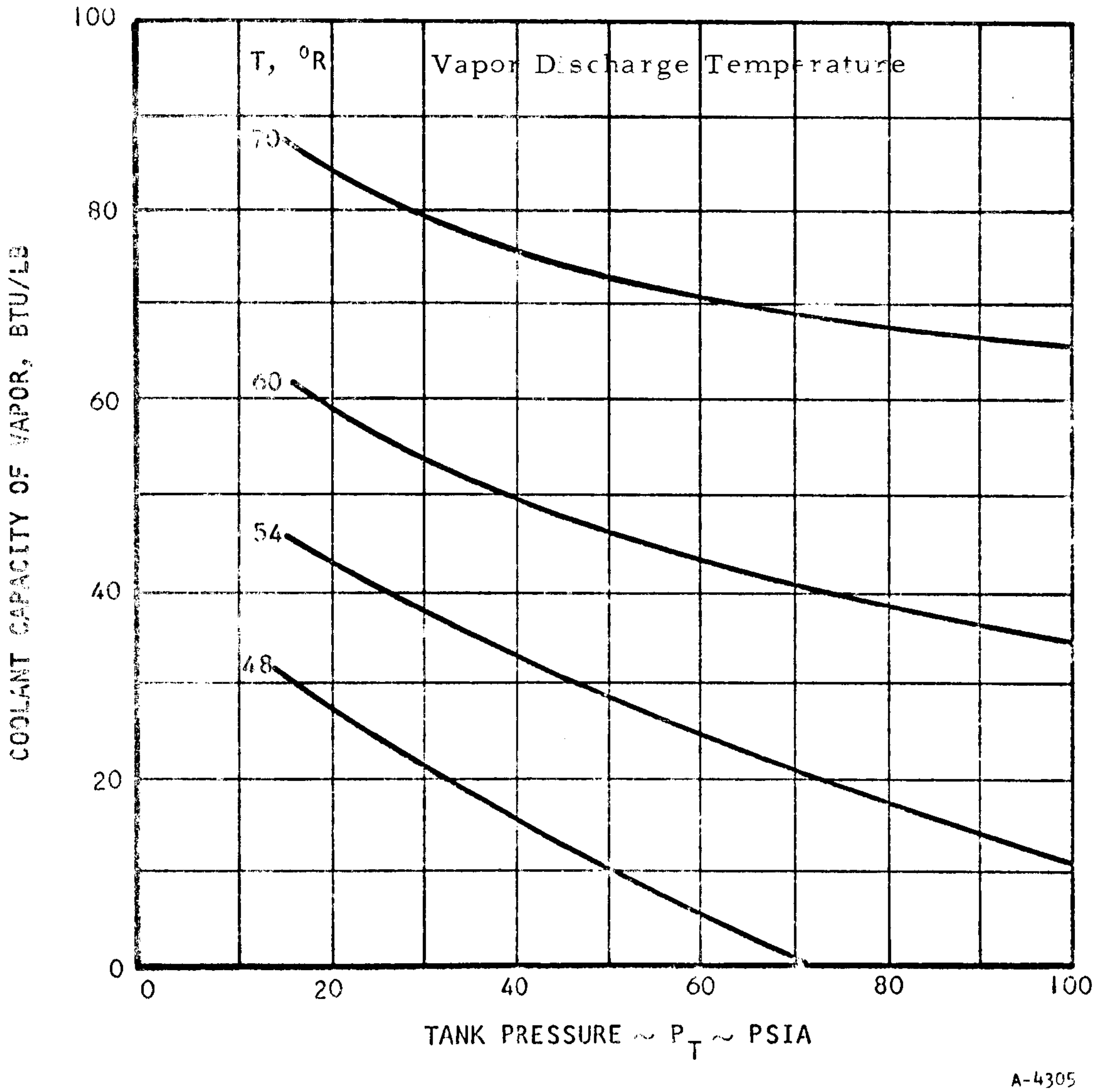

Figure 3. Effect of Catch Tank Pressure and Heat Exchanger Discharge Temperature on Vapor Cooling Capacity

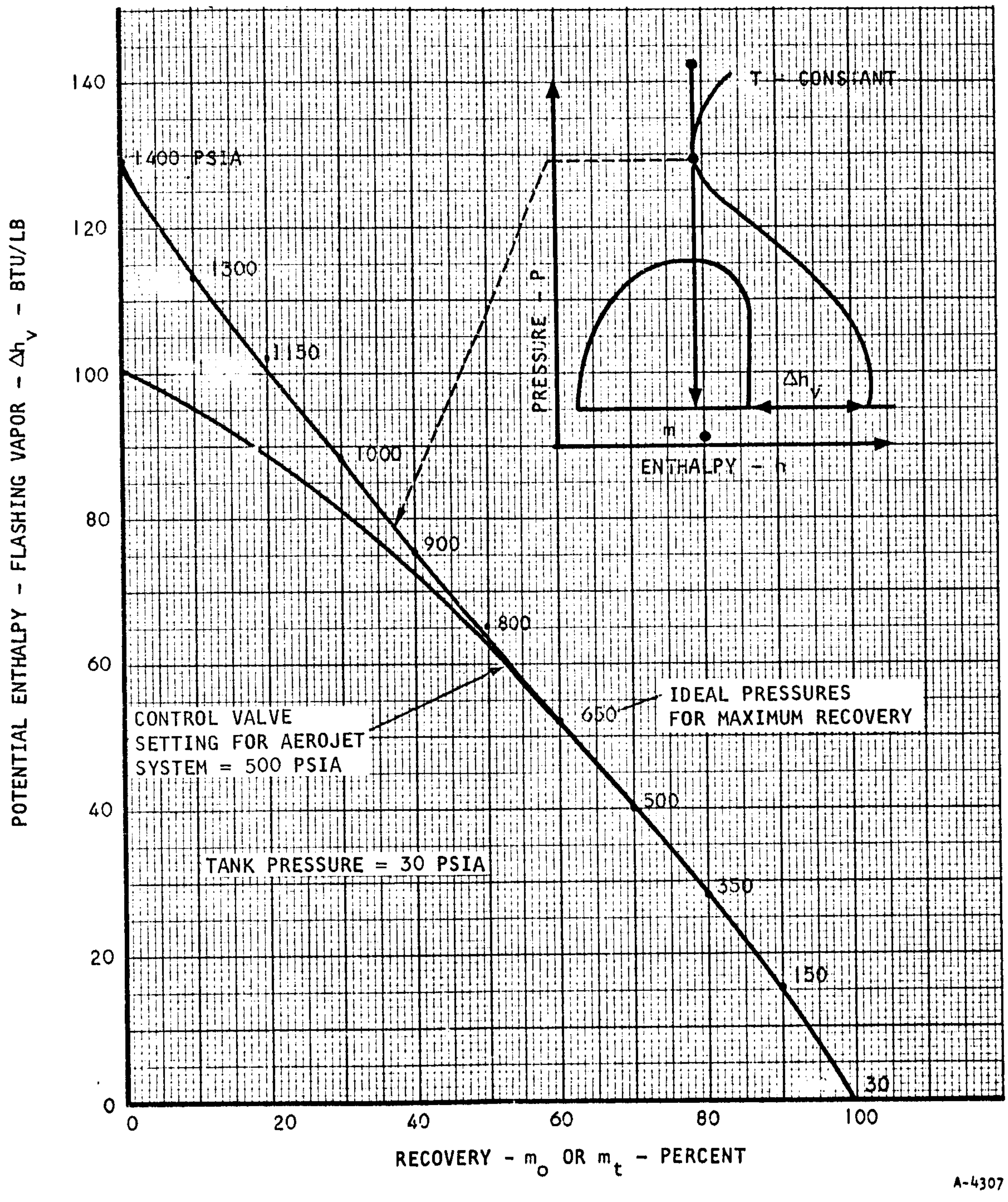

Figure 4. Enthalpy of Flashing Vapor Relative to High Pressure Cryogenic Hydrogen

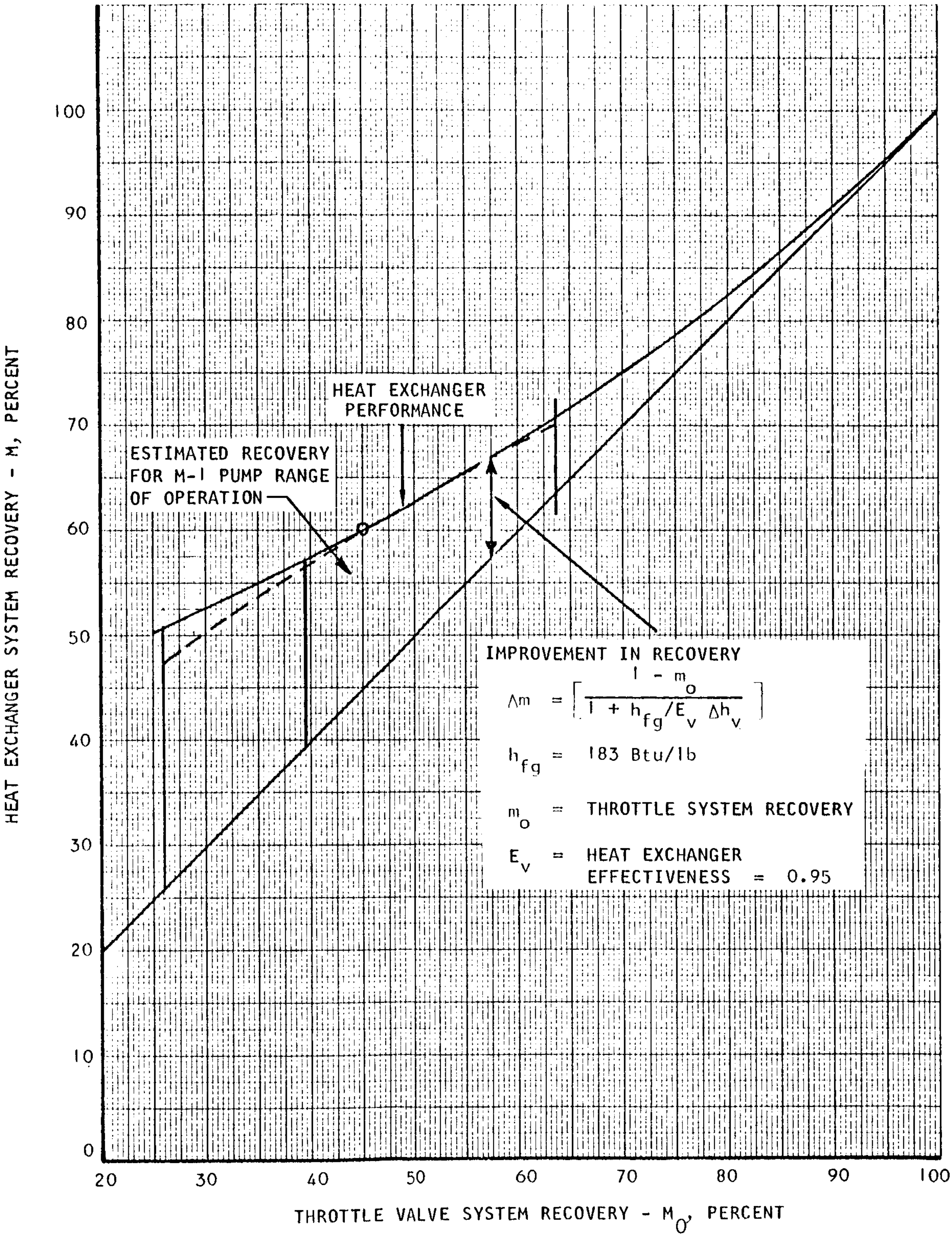

Figure 5. Heat Exchanger Recovery System Performance Compared to Simple Throttling System Recovery

the system recovery is 60 percent, or 15 percent above the recovery of the throttling system. This increase represents an improvement of 15/45, or 33 1/3 percent, over the basic system.

d. Effect of Liquid Entrainment

An added feature of the heat exchanger system is the evaporation and elimination of liquid from the vent gas. This results in an improvement in recovery which is directly proportional to the amount of entrained liquid.

Near the vent gas inlet, the heat exchanger acts simply as a subcooler where the net energy exchange is due to the heat of evaporation of the liquid. This energy exchange only brings the recovery up to the amount that would occur if there were no liquid entrainment. Then the remaining portion of the exchanger performs in the manner described previously. Obviously, this performance will be less efficient in the sense that the full cooling potential of the resulting vapor is not utilized. By increasing the surface area and/or increasing the vapor pressure drop through the exchanger so that a high effectiveness is maintained, the full potential recovery of the heat exchanger method can be obtained with liquid entrainment in the vent gas. This is shown analytically in Appendix A.

3. Turbine Recovery System

The greatest improvement in recovery is obtained by a turbine which extracts work from the high energy fluid. Figure 1D shows the system diagram. The turbine is located near the test pump primarily to minimize the length of high pressure lines. Unlike conventional turbomachinery wherein there are large changes in density, the location is not critical. However, an optimum does exist, and this optimum does exist, and this optimum is more a function of economic factors than pressure drops fore and aft of the turbine in the transfer lines and catch tanks.

Referring to Figure 2, the turbine energy extraction, Δh_t, is a function of the available pressure differential after the appropriate deductions have been made for fluid transfer, changes in elevation, and single-phase flow pressure level requirements. This extraction may be expressed simply by

$$\Delta h_t = \frac{\Delta P_t \, \eta_t}{J \rho_{av}}$$

where ΔP_t = pressure drop across turbine

ρ_{av} = average density

η_t = adiabatic efficiency

or, may be obtained from the Mollier diagrams of thermodynamic properties. Then the turbine system recovery may be expressed

$$m_t = 1 - \frac{(h - h_o) - \Delta h_t}{h_{fg}}$$

Subtracting the simple system recovery, the net improvement due to the turbine is

$$m_t - m_o = \frac{\Delta h_t}{h_{fg}} = \frac{\Delta P_t \eta_t}{J \rho_{av} h_{fg}}$$

a. <u>Effect of Efficiency</u>

The above expression indicates a direct proportionality between recovery and efficiency, where the proportionality constant is the ideal work extraction divided by the heat of vaporization of the catch tank liquid,

Thus

$$\frac{d(m_t - m_o)}{d\eta_t} = \frac{\Delta P_t}{J \rho_{av} h_{fg}}$$

b. <u>Effect of Pressure Drop</u>

These relations also indicate the relative importance of pressure drops in the lines, since line friction losses and restrictions reduce the pressure drop available for the turbine. The loss in recovery, therefore, is dependent upon the attainable turbine efficiencies over the usable pressure drop. For the range of pressures and temperatures of the M-1 test pump, the penalties for high pressure or low pressure restrictions are nearly equal so that an incremental loss in recovery, Δm_L, can be expressed as

$$\Delta m_L = \frac{\Delta P_L \, \eta_t}{\rho_{av} J \, h_{fg}}$$

where ΔP_L = line pressure drop

η_t = attainable turbine efficiency

ρ_{av} = average density over the available pressure drop to the turbine.

c. Turbine Effect on Liquid Entrainment

The substantial increase in recovery will mean a corresponding reduction in vent gas flow rate and a reduced turbulence within the tank, particularly in the initial part of the test run. At the end of the run, a fuller tank will result in a high turbulence, and probably a high percentage entrainment loss. Therefore, the primary effect will be a lower entrainment loss due to the greatly reduced flow rate of vent gas.

4. Combination Turbine and Heat Exchanger System

The combination of a turbine and heat exchanger will result in a small improvement over the turbine system alone, particularly for the lower recoveries obtained for high energy, or off-design point conditions. A heat exchanger downstream of the turbine (Figure 1E) would extract an additional recovery,

$$m - m_t = \frac{1 - m_t}{1 + \dfrac{h_{fg}}{E_v \Delta h_{vt}}}$$

where m_t is the turbine recovery and $\Delta h_{vt} = f(m_t)$, Figure 2.

Clearly, if the turbine recovery m_t, is high, the heat exchanger contribution, $(1 - m_t)$, is small.

Again, this system is subject to the same operating pressure constraints as the heat exchanger system, and low tank pressures and vent line pressure drops are required for attaining the above performance.

If the heat exchanger is located upstream of the turbine, (Figure 1F) the improvement is expressed:

$$m - m_t = \frac{(1 - m_o) - \Delta h_t / h_{fg}}{1 + h_{fg}/E_v \Delta h_{vo}}$$

where $\Delta h_{vo} = f(m_o)$, Figure 2.

Since $m_t = m_o + \Delta h_t/h_{fg}$, the two expressions are nearly the same except for the value of Δh_{vt} and Δh_{vo}, and a small change in the available work Δh_t, for the two conditions. Since Δh_{vo} is considerably greater than Δh_{vt}, a greater recovery is possible with the upstream position. However, this position may require a long, large-diameter vapor line from the recovery tank which will incur a severe heat leakage penalty and thereby reduce the effectiveness of the vapor for subcooling the fluid into the turbine. Either the maintenance of this line at cryogenic temperatures or the transient heat capacity would add more heat than the vapor can absorb. Therefore, this system will not perform as indicated in installations where the distances between the basic components are as great as that of the Aerojet test stand.

5. Typical Performance Values

Attainable turbine efficiencies are dependent upon specific speed relations as discussed in the next section. For the M-1 pump flow rates and pressures, an efficiency of 90 percent appears possible. Allowing 200 psi for pressure drops and single-phase flow pressure requirements in the return line, the potential increase in recovery for a turbine is

$$m_t - m_o = \frac{1600 \times 1.44 \times .90}{778 \times 4.3 \times 1.84} = 34 \text{ percent}$$

The improvement over the throttling system is then $\frac{34}{45}$, or 75 percent.

The relation of efficiency to recovery is then

$$\frac{d \Delta m_t}{d\eta_t} = 0.38 \quad \text{percent recovery/percent efficiency}$$

Similarly, the loss of recovery due to pressure drop is

$$\frac{\Delta m_L}{\Delta P_L} = 0.021 \ \frac{\text{percent}}{\text{psi}} \text{ or } 2.1 \ \frac{\text{percent}}{100 \text{ psi}}$$

When these values are converted into total flows of liquid hydrogen and related to the cost per gallon, the basic economic implications of the above recovery methods can be ascertained.

Based upon a current cost estimate of $0.50 per gallon for liquid hydrogen, the potential or gross savings factors of the above parameters and components are summarized in the following table.

	Total gallons LH_2		126,000,000
	Cost of LH_2 (at $0.50 per gallon)	$	63,000,000
(1)	Value of one percent recovery	$	630,000
(2)	Value of each percent turbine efficiency	$	238,000
(3)	Value of each 200 psi pressure loss in lines of turbine system	$	1,400,000
(4)	Value of heat exchanger recovery at an effectiveness of 0.95	$	9,500,000
(5)	Value of turbine recovery, 90 percent efficiency, 230 psi line pressure drops	$	21,400,000

These values show the relative importance of some of the basic parameters which govern the economic aspects of system design. From these (gross) values must be deducted the cost of the recovery system and its maintenance. These costs are evaluated in the following sections of the report.

Figures 6 through 10 present charts which are based upon the relations presented above. Figure 6 shows the recovery and dollar value of the simple recovery system for various pump pressures and efficiencies. The dotted lines, which indicate the effect of tank pressure, show the apparent recovery during a run as compared to the actual recovery as a result of boiloff after pressures have receded to the final "use" pressure.

Figure 7 shows the improvement in recovery of the heat exchanger system. Total recovery is obtained by adding the corresponding values of the simple system from Figure 6. This chart is based upon a tank pressure of 30 psia during operation. A subsequent decay to 15 psia would reduce the performance as indicated in Figure 6. These values do not include liquid entrainment losses, which may be considered fully recoverable by the heat exchanger system. Thus, an entrainment loss of 5 percent will add this amount to the recovery shown in Figure 7.

Figure 8 presents a similar chart for the turbine-absorber system based on an 80 percent pump efficiency. Figures 9 and 10 present similar values based on pump efficiencies of 70 and 60 percent, respectively. The savings shown are from the improvement in recovery over the simple system. Total recovery is obtained from the sum of Figures 6, 7, and the appropriate turbine curve (either Figure 8, 9 or 10).

With respect to entrainment losses for the turbine system, this loss may be considered reduced by an amount which is proportional to the total change in recovery. Thus, if the simple system yields 40 percent recovery with an entrainment loss of 10 percent, and a turbine system has a total recovery of 80 percent, the vapor flow rate has been reduced from 60 percent to 20 percent, or one third. The corresponding entrainment loss would be 3 1/3 percent and the total recovery about 77 percent. Thus, the improvement due to the turbine would be 47 percent instead of 40 percent as indicated by the charts.

6. Cooldown Losses

The amount of liquid hydrogen that would boil off as a result of contact with normal temperature equipment is very high, and varies with the degree of sensible cooling that is done by the vapor before its escape into the atmosphere. The maximum boiloff, as exemplified by the immersion of a solid into a pool of liquid is 0.35 lb/lb for aluminum

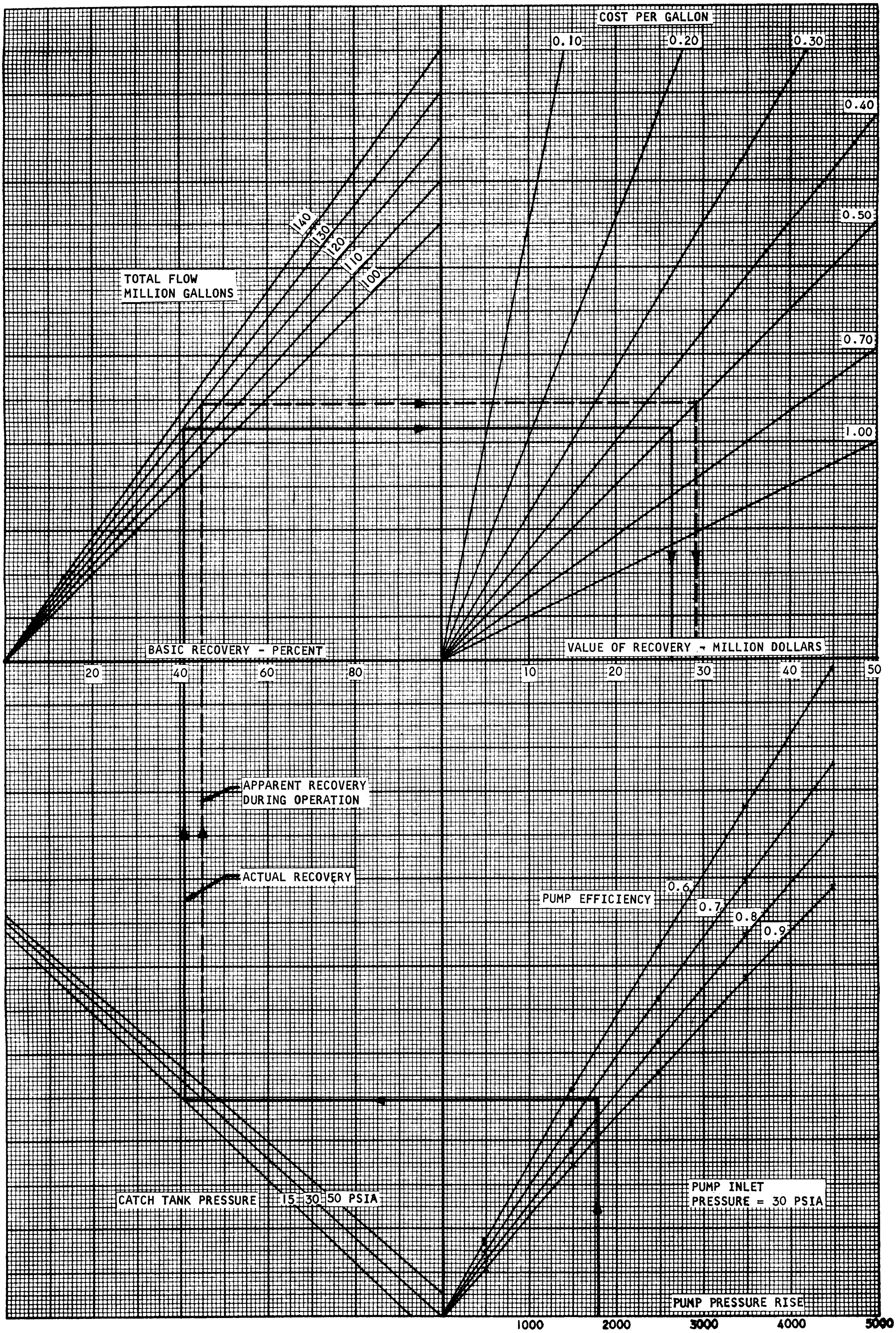

Figure 6. Simple Throttling Recovery System

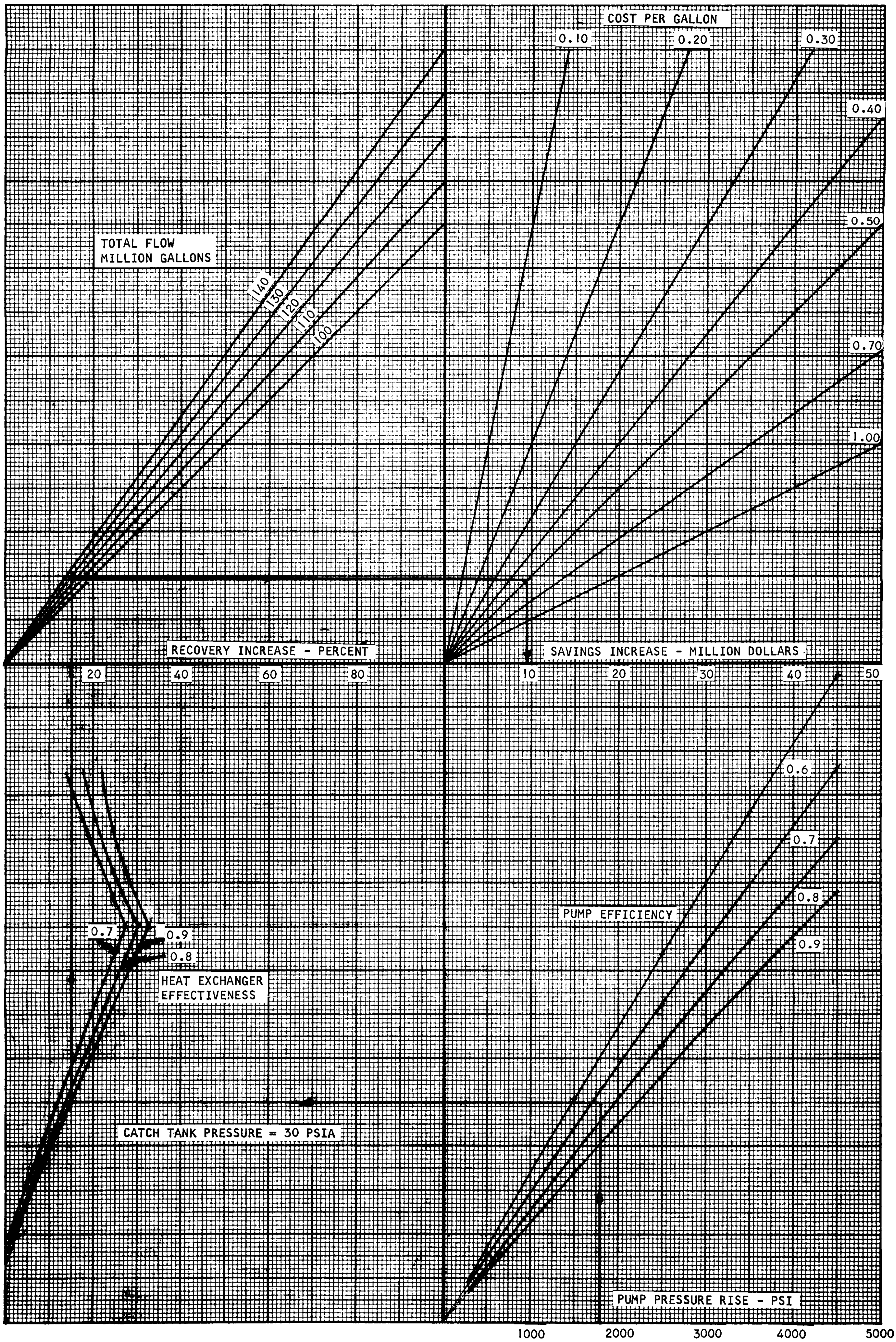

Figure 7. Heat Exchanger Recovery System - Increase in Recovery Over Basic System

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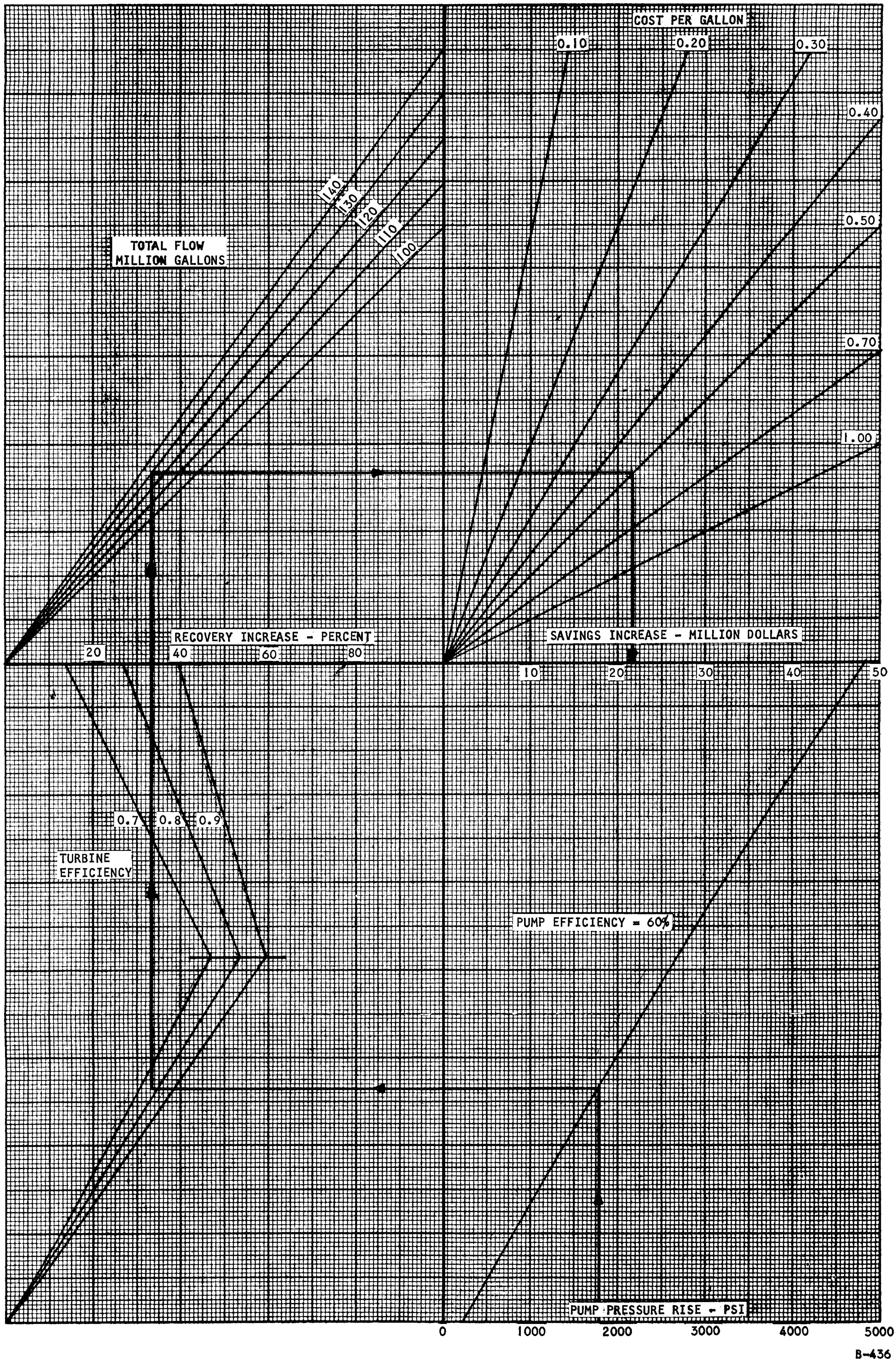

Figure 8. Turbine Power Absorber Recovery System as Compared to Basic System

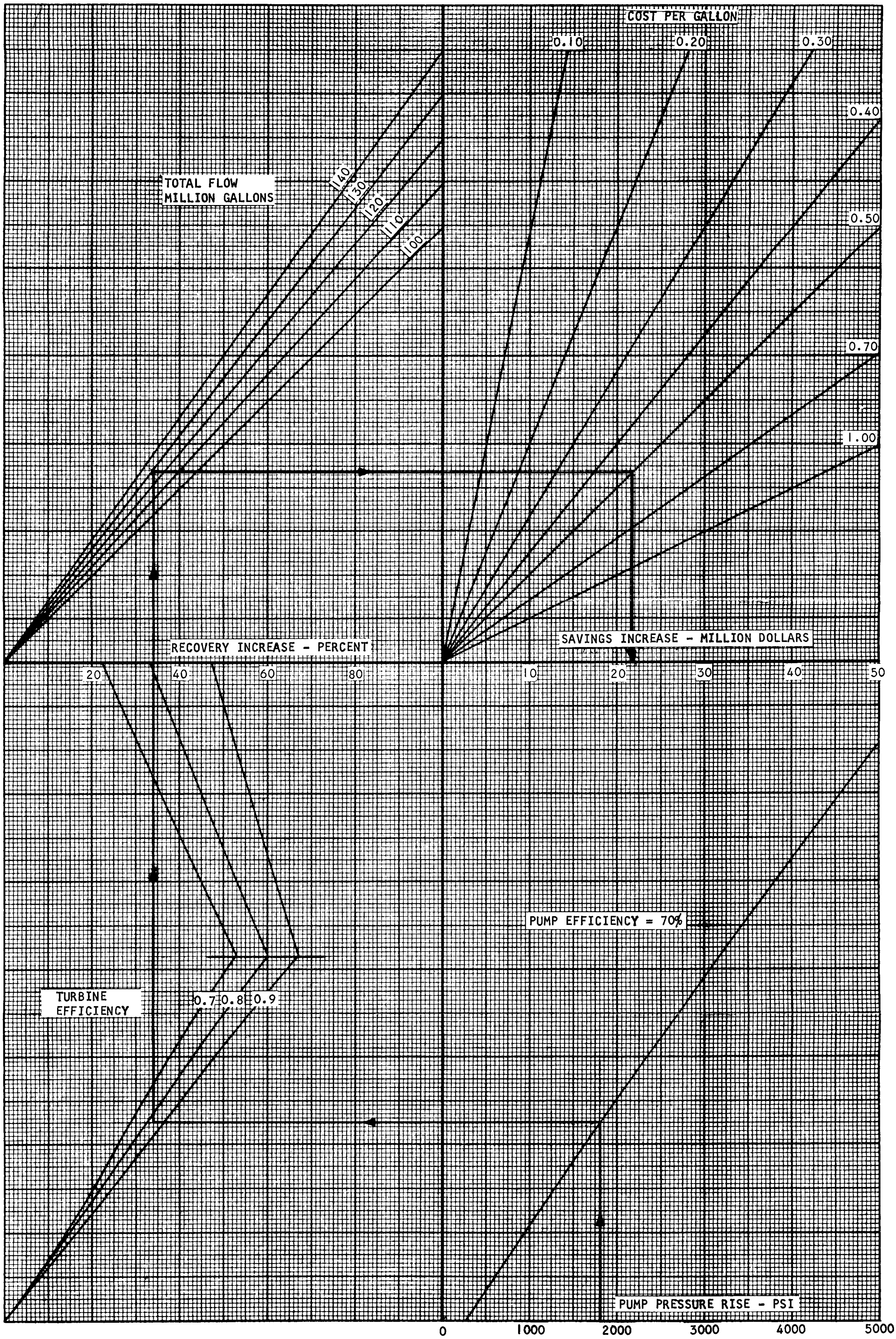

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Figure 9. Turbine Power Absorber Recovery System as Compared to Basic System

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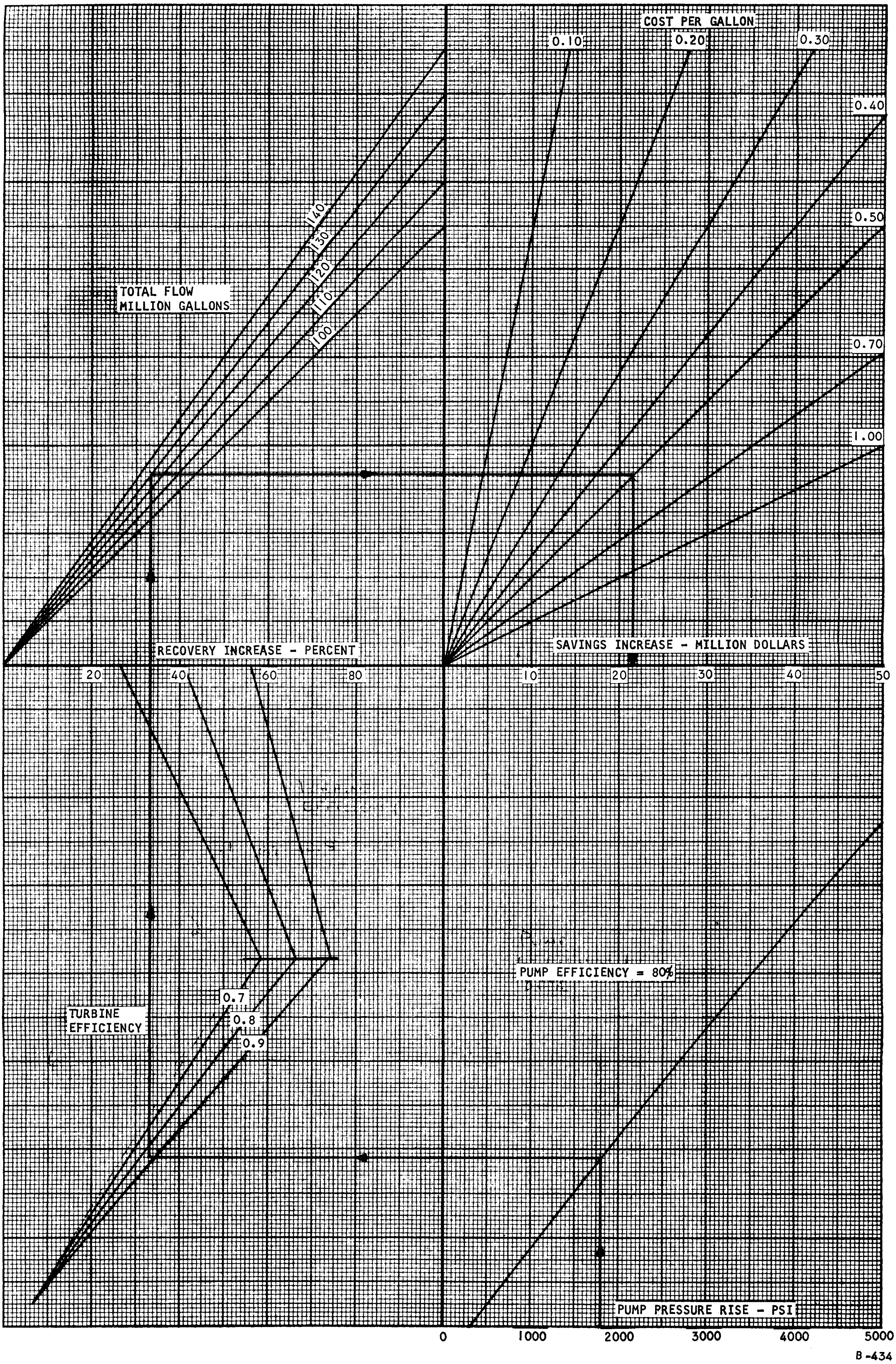

Figure 10. Turbine Power Absorber Recovery System as Compared to Basic System

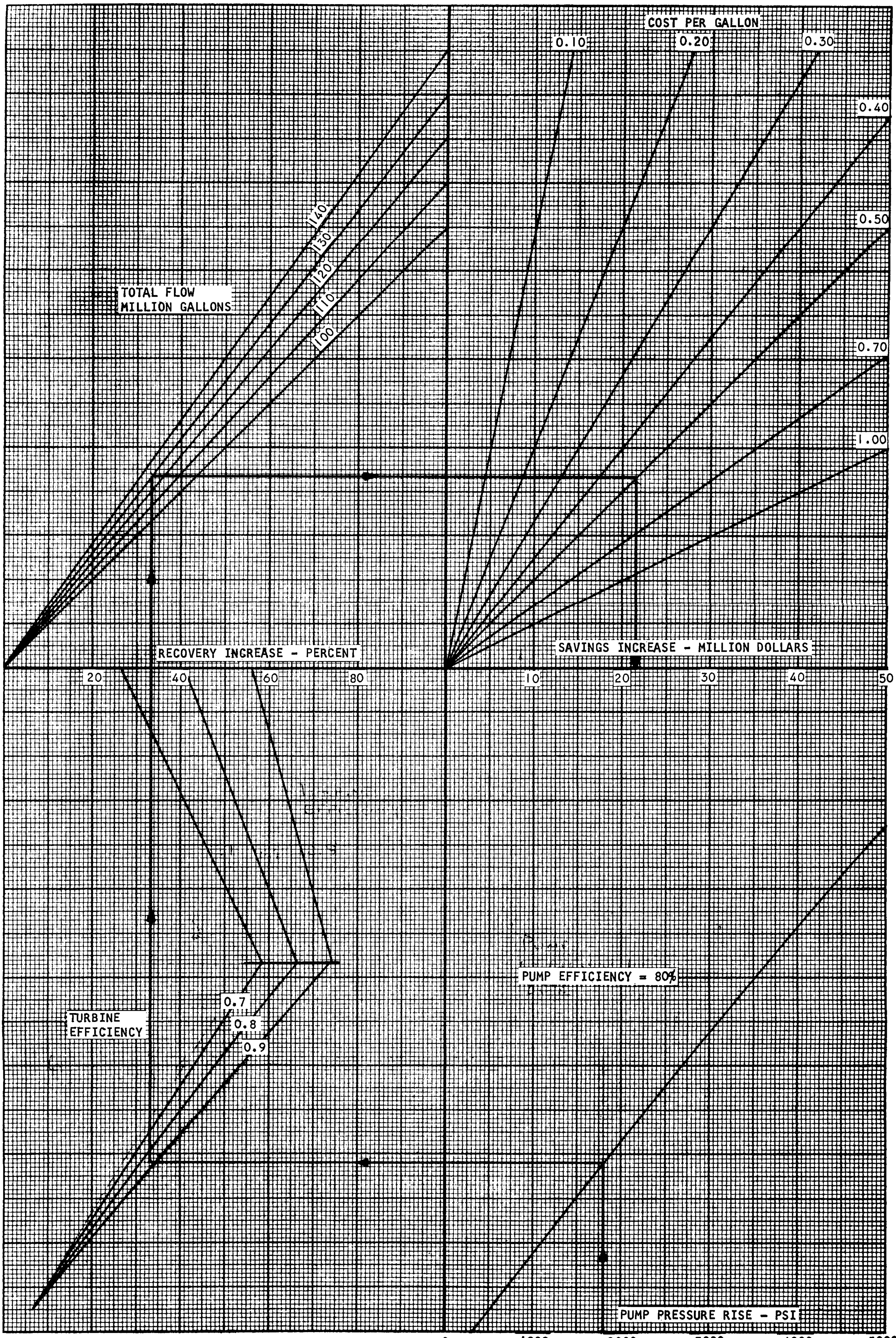

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Figure 10. Turbine Power Absorber Recovery System as Compared to Basic System

and 0.18 lb/lb for steel. Here, only the latent heat of evaporation cools the material. The minimum boiloff is obtained when the vapor is not released until its temperature and the material temperature are the same, which is the type of cooling that may be obtained in a long pipeline or heat exchanger at low flow rates. The minimum liquid hydrogen required is 0.07 lb/lb and 0.036 lb/lb for aluminum and steel, respectively. The difference between minimum and maximum requirements are obviously substantial, the ratio being roughly 5 to 1.

The inner pipe of the Aerojet return line weighs about 45,000 lb and would incur a maximum boiloff of 8100 lb of liquid hydrogen during startup from a warm condition. This amount is roughly 4-1/2 to 10 percent of the total pump flow per run. If a small amount of LH_2 were admitted prior to each run, the minimum boiloff would be attained, or 1620 lb/run.

The costs of these cooldowns, at 50 cents/gallon, would be about \$7000 maximum and \$1400 minimum, per run. For 84 test runs per year, these costs are \$588,000 and \$117,500, and the five-year totals are \$2,940,000 and \$588,000, respectively. For a five-ton turbine which is made of steel, the range is \$645,000 to \$129,000 for the five-year period. The 25-ton heat exchanger, made of aluminum, would incur the minimum cooldown losses due to a high surface-to-weight ratio and would be \$1,250,000 for the test period.

Clearly, methods of reducing the cooldown losses are desirable, provided that costs do not exceed the savings. The simplest method is to reduce equipment temperatures by circulating the storage tank boiloff vapors through the line and equipment between runs. This vapor is essentially free and has a generally adequate cooling capacity if the fluid levels in the supply and catch tank are above a certain minimum, and if the heat leakage into the equipment is minimized. For short lines, a foam-type insulation may be adequate, whereas for the Aerojet facility, a superevacuated-type insulation can readily be justified.

Basic Parameters Affecting Cooldown Losses

At cryogenic temperatures, the cooldown losses are reduced in two ways; a lower temperature drop, and a reduced specific heat of the metal. The specific heats of both aluminum and steel at 200°R are about one-half of their normal temperature values. In addition, the effective conductivity of the cryogenic insulation, particularly foil-wrapped vacuum jacketing, is lower so that heat leakage is not linearly proportional to the temperature difference. Thus, at a reduced temperature of 100-200^{o}R, the maintenance cooling does not require a proportional increase in boiloff liquid or vapor flow rates to maintain the low temperatures. As shown in Appendix B, optimum maintenance temperatures exist for various types of equipment, and these temperatures are dependent upon the surface area weight ratio times the conductivity of the insulation. In other words, the cooldown penalties of cryogenic equipment are very similar to the drag penalties of airborne equipment. Weight and volume, which are generally not critical for ground installations, become important parameters in cryogenic installations.

With very large scale structures, such as the E-2 test facility, an additional factor arises which lessens the cooldown loss and which complicates the problem of analysis. This factor is the large heat capacity and the resultant thermal lags that are inherent in well insulated, large scale cryogenic equipment. On the basis of the test schedules for the M-1 pump, steady-stage conditions will not be attained between runs, so that cooldown requirements must be determined from the transient thermal response characteristics of the system, rather than from steady-state relations.

Lumped Parameter Time Constants

Lumped parameter time constants are useful for obtaining an approximation of the rate of temperature change of an insulated material as a result of a heat input which is proportional to the temperature difference. A simple heat balance on a slab of material having a weight, W, and a surface area, A, is

$$WCdT = UA_s \, (T_a - T) \, d\theta$$

The first order time constant is

$$\tau = \frac{WC}{UA_s} = \frac{\rho tC}{k/x}, \text{ approximately}$$

The initial rate of change is

$$dt/_{d\theta} = \frac{T_a \; - T_o}{\tau}$$

where C = specific heat of the material (at LH_2 temperature)

k/x = conductance of insulation

ρ = density of the material

t = thickness

T_a = ambient temperature

T = material temperature

T_o = initial material temperature

For a large storage tank with an inner wall of steel, one inch thick, and insulated with two ft of evacuated Perlite, a typical time constant is:

$$\tau = \frac{500 \times 1/12 \times .05}{.0007/2}$$

$$= 6000 \text{ hours, or about 8 months}$$

The initial rate of change would be $\Delta T/_\tau$ or $\frac{484^\circ}{6000} \approx .08^\circ$ per hr, or less than 2 deg per day.

Pipe line temperature change rates would be greater due to less insulation and thinner walls. For the Aerojet configuration where $k/x = .009$ Btu/hr-ft^2 - $^\circ$F and the 18" pipe wall thickness is 0.438 in.

$$\tau = 140 \text{ hr or } 5.8 \text{ days}$$

The initial rate of change would be

$$\frac{484}{5.8} = 43^\circ/\text{day}$$

The warmup process of a turbine between runs is considerably more complex; however, certain simplifications are readily apparent from a consideration of the primary resistances to heat flow. As with the tank and pipe line, the major resistance is the insulation, which will be a vacuum, powder-filled jacket. The resistance through the housing and into the blades is negligible. Between the blades and rotor structure, the primary resistance is convective hydrogen, which is low at cryogenic temperatures. However, the surface area of the blading is high so that the overall resistance between the rotor and stator is small compared to that of the insulation. Therefore, little error is incurred if the entire mass and structure of the turbine is considered a lumped parameter for which the first order time constant may be written

$$\tau = \frac{WC}{k/x\, A_s}$$

where A_s = 100 sq ft

W = 10,000 lb

C = .05 Btu/lb°R at cryogenic temperatures

k/x = .010, includes support conductivities

$$\tau = \frac{10,000 \times .05}{.01 \times 100} = 500 \text{ hr}$$

Thus, the initial warmup rate of the turbine will be about

$$\frac{dT}{d\theta} = \frac{\Delta T}{\tau} = 1^{\circ}/\text{hr}$$

or 24° per day

A similar approach may be taken with respect to the heat exchanger response time. Here, the outer shell is connected thermally to the tubing bundle by many baffle and tubing support structures, side walls, and by free convection conductances. The thermal resistance between the inner mass and outer shell is small compared to the resistance across the insulation, so that the lumped parameter approach is applicable. Since the structure is aluminum, the following values apply.

W = 50,000 lb

C = 0.10 Btu/lb°R at cryogenic temperatures

A_s = 800 sq ft

k/x = .005 (x = 6 in. supports included)

$$= \frac{50,000 \times .10}{.005 \times 800} = 1250 \text{ hr}$$

and $$\frac{dT}{d\theta} = \frac{484}{1250} \times 24 = 9.3^{\circ}/\text{day}$$

These approximations provide a preliminary insight into the warmup characteristics of the system and the cooldown losses that may be incurred by the various components of the recovery system during startup. After the initial cooldown, the boiloff vapors from the supply and catch tanks will vary with liquid levels in the tanks but will not vary greatly in temperature as a result of the large thermal lags in the tank structure. As the vapor passes into the lines the temperature rise will be near that of the pipe walls, which means that the vapor would enter the turbine section at a higher temperature than the turbine, and, in effect, would increase the turbine temperature faster than heat leakage from the outside. Thus it would appear to be preferable to bypass the turbine between runs to minimize its cooldown penalty. However, on an overall system basis, this concept may or may not be valid due to the variation in specific heats and insulation conductances at liquid hydrogen temperatures. If the average pipe line temperature is lowered as a result of the transient heat capacity of the turbine to the extent that the average specific heats and conductances of the insulation of the complete system are lowered, then the turbine should be included in the vent circuit.

Obviously, detailed analysis of particular systems are necessary for the evaluation of these inter-relations. Fortunately, these effects on the costs of the heat exchanger and turbine-absorber recovery systems for the Aerojet facility are relatively minor, so that elementary approximations are satisfactory at this time. As indicated by the time constants above, the turbine temperature rise in 4 days would be about $96^{\circ} + 42^{\circ}$ or 138°R. From Appendix B, the cooldown loss ratio, σ, is .0042 (min) and .008 (max). For the turbine, which has a weight of about 10,000 lb, this loss is 42 to 80 lb per run, or 3500 to 6400 lb of liquid hydrogen per year.

Similarily, the heat exchanger temperature rise would be 4 x 9.3 = 37°, or a rise of 79°R before each run. The cooldown loss factor for aluminum is $\sigma = .0015$ min and .002 max. At a weight of 50,000 lb, the cooldown loss would be near the minimum of 75 lb/run or 64 lb/year.

When these values are compared to those indicated for the uninsulated components, it is clearly evident that substantial savings result from insulating and maintaining low temperatures.

II. E-2 TEST STAND

The design and performance of a recovery system is dependent upon the pressure levels and flow rates in the system and the pressure drops necessary for movement of the fluid and vapors. Also, the dynamics of the system and the required controls are related to the mass of fluid in the lines and the response rates of the control elements.

With respect to the E-2 test stand, the existing tankage, lines, and control valves represent certain constraints which have an important bearing onthe design of components which must be integrated into the system.

Figure 11 shows a line diagram of the E-2 test area recovery line, tank, and vent system. Figure 12 shows the location, routing, and principle dimensions of the lines, and the control elements at the catch tank.

An important consideration in the design of a turbine power absorber is the range of operation of the test pump and the amount of test time or liquid hydrogen that is expended in off-design point testing. Figure 13 shows the operation envelope for the M-1 pump and Figure 14 gives the time duration of the tests for a period of three years. The upper curve of Figure 14 shows the amount of time spent per quarter on design point and off-design point tests, and the lower curves show the accumulated test time for the three-year period. It is evident that the off-design point tests will require less than 10 percent of the total time, and approximately 10 percent of the total liquid hydrogen consumption.

Other data pertinent to the study which was requested from Aerojet is summarized as follows:

Heat Influx from Pump to Catch Tank

10, 500 Btu/hr

Pressure Drop K Factors

a. Control valve to orifice: K = 4. 4

Line diameter: 12 in.

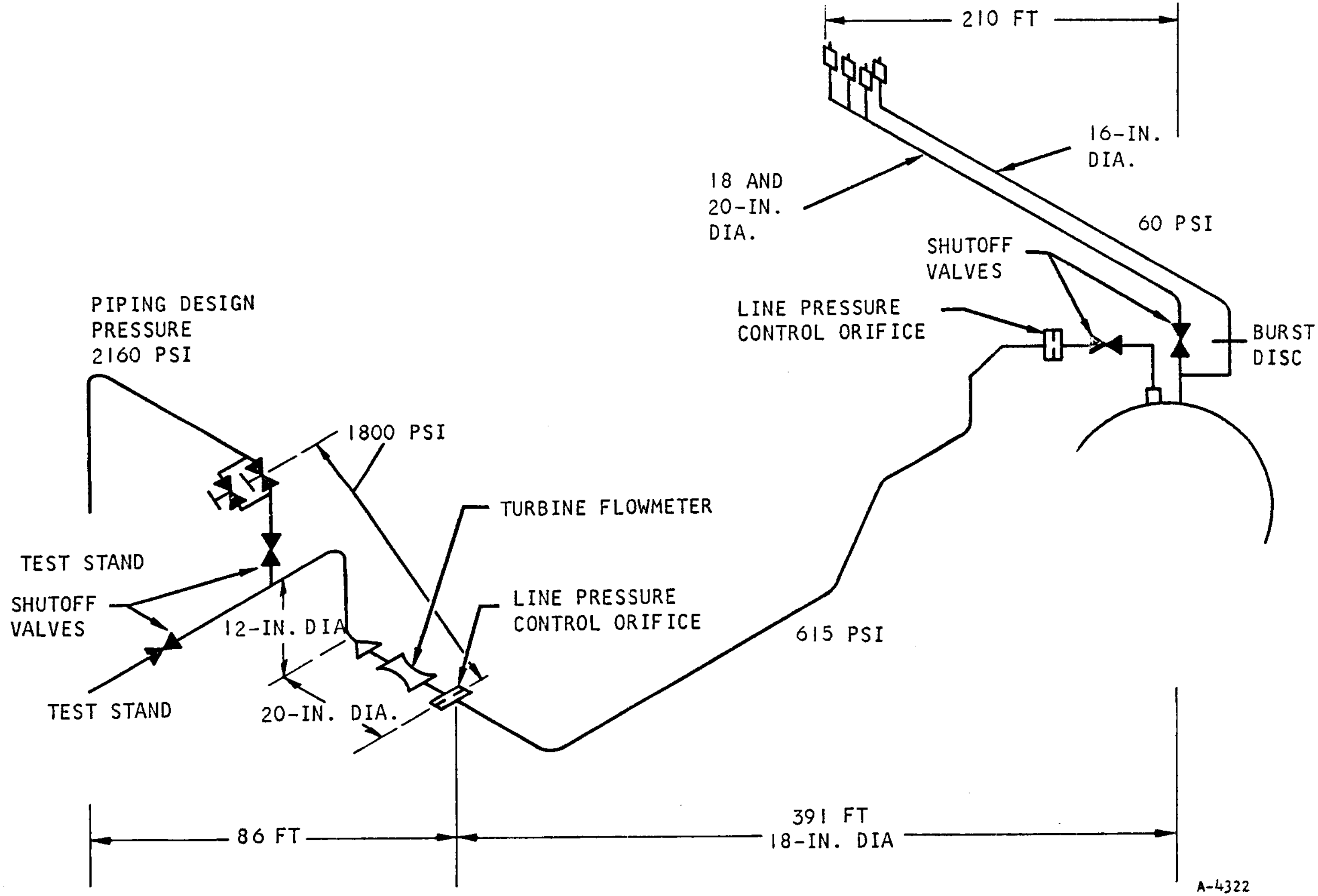

Figure 11. Liquid Hydrogen Return Line, E-1 Test Stand
Aerojet-General Corporation, Sacramento, California

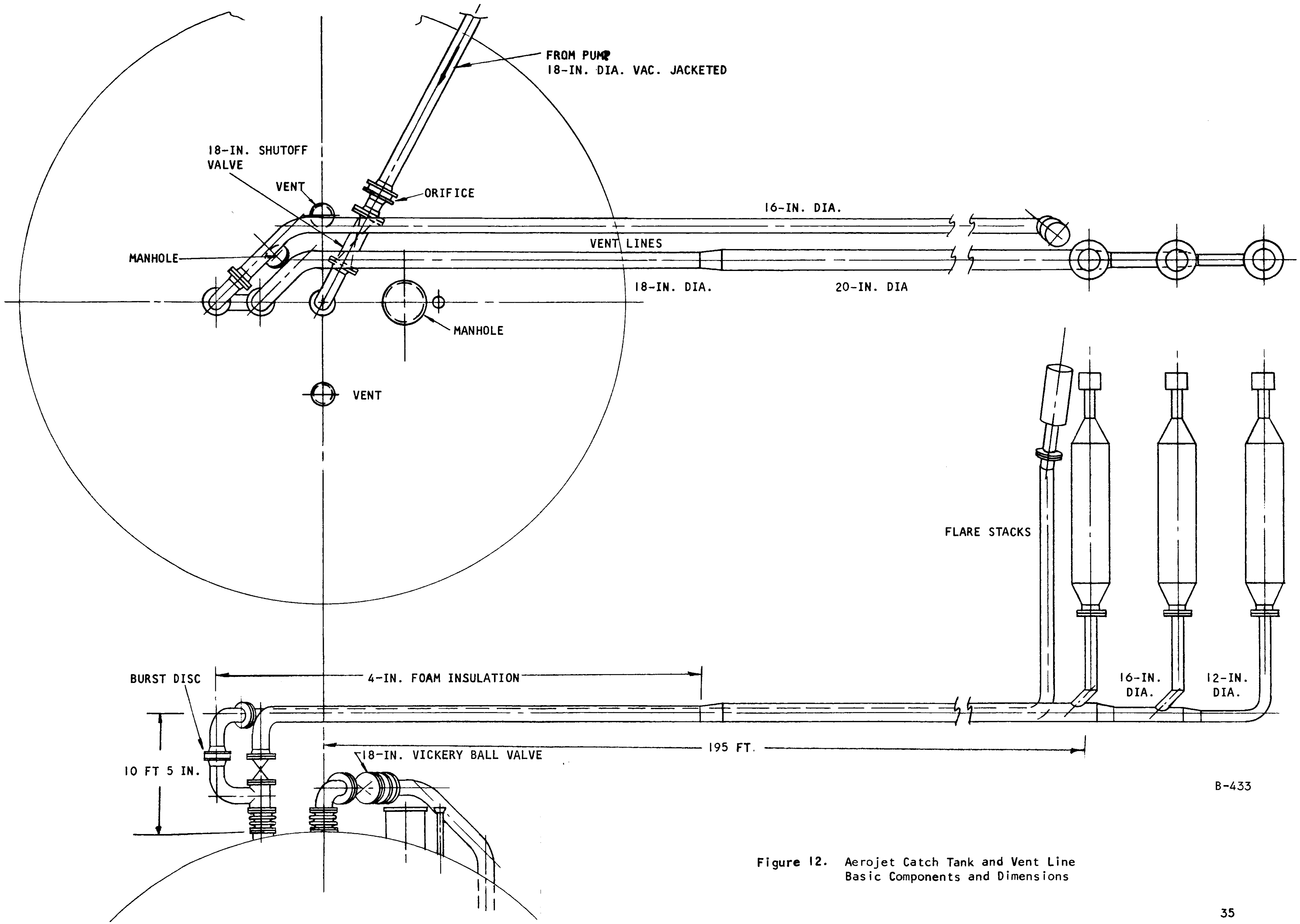

Figure 12. Aerojet Catch Tank and Vent Line Basic Components and Dimensions

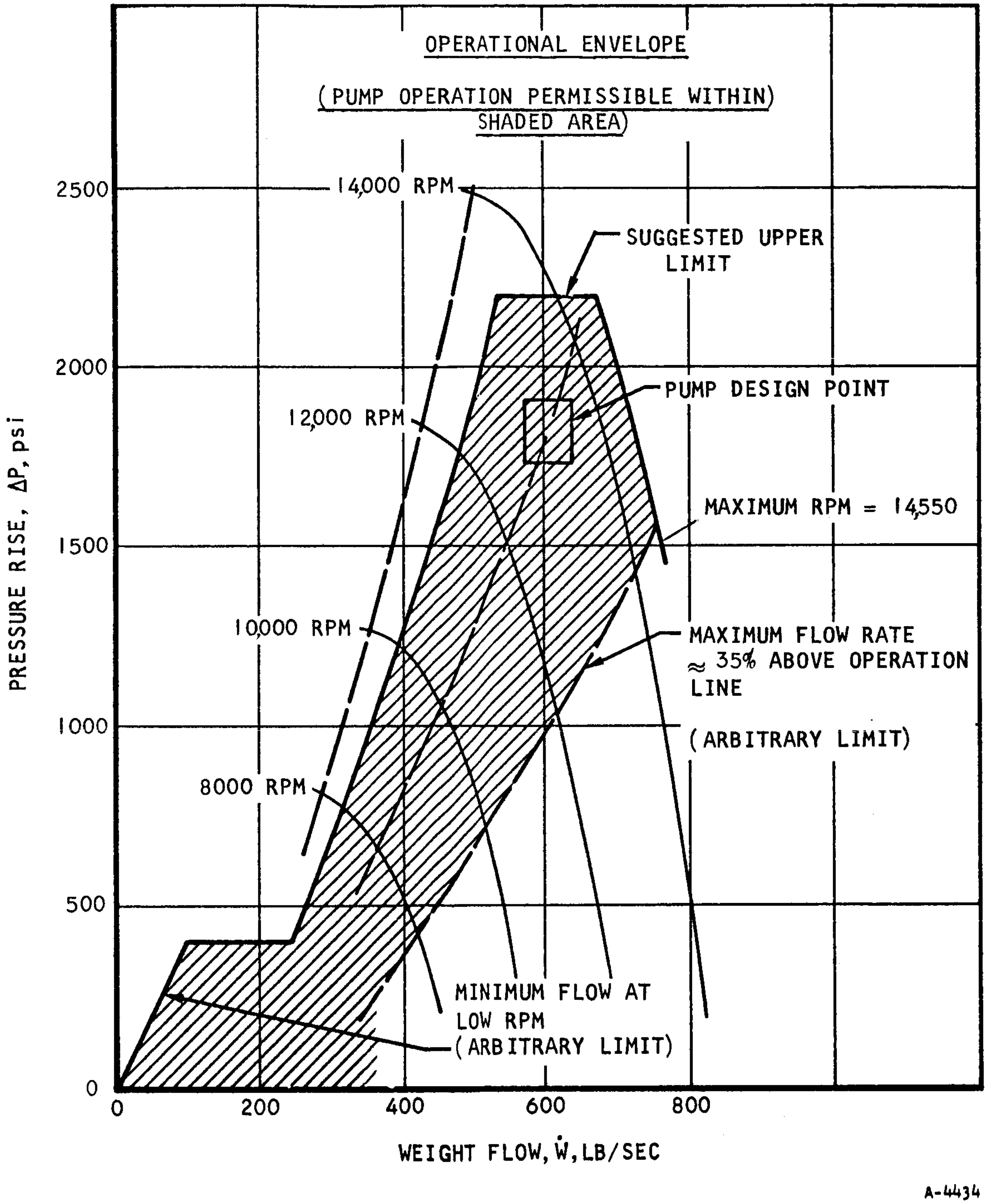

Figure 13. Model II Fuel TPA

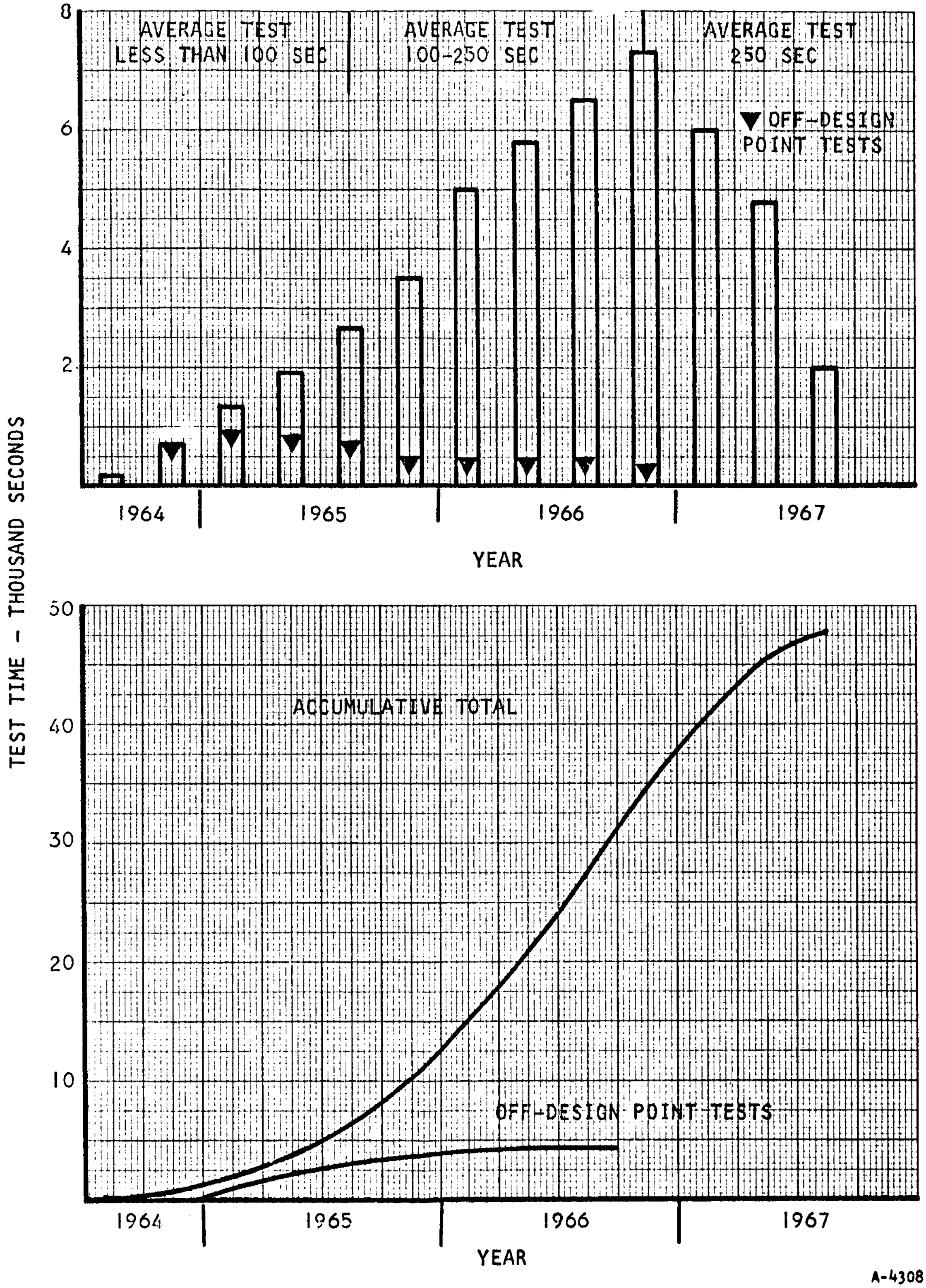

Figure 14. M-1 Pump Test Schedule

b. Orifice to Catch Tank: K = 5.8

$$K = fl/d = .014\ l/d$$

Line diameter: 18 in.

Vent Flow Rates and Flare Velocity

The minimum velocity for the flares can be determined from the boiloff of LH_2 from V-E2 which may be as low as 110 lb in any 24-hour period. The working maximum is expected to be approximately 18 lb/sec at 60°R. The maximum condition for which the flare is designed is 120 lb/sec at 37°R. This flow will result in an exit velocity of approximately 1100 fps. Pressure drop work is currently done using a composite K factor of 5. Exit diameter is 17.75 inches.

Load Capacity of Vent Line Supports

Preliminary analysis indicates that the vent line support towers could carry the weight of an additional 24-in. vent line without any major structural changes. All main columns are adequate.

Cooldown Conditions

The vent gases from the supply and catch tanks are circulated through the transfer lines between runs to maintain low temperatures and minimized cooldown requirements. The total flow rate is about 1000 gallons per day. The cost of the present recovery system was also requested for the purpose of comparison with the estimated cost of the equipment to be added. The approximate total is $1,435,000 of which $245,000 is for 7 valves and $200,000 is for controls, purge and deluge systems and instrumentation.

III. TURBINE DESIGN STUDIES

From the previous studies, it is evident that the important parameter for a turbine recovery system is a high stage efficiency. Since each percentage point of recovery is worth about $630,000 in terms of the cost of liquid hydrogen, each percentage point in efficiency is equal to 0.37 x 630,000 or $232,000. Therefore, a design which would produce the maximum attainable efficiency at a reasonable cost in money and time appeared to be the proper approach to the problem.

From the specific speed standpoint, the head-capacity-relations indicated that an efficiency of 88 to 92 percent was attainable in either a single radial machine, or in a multistage axial machine.

The single-stage radial machine would have the added advantage of providing a high off-design point efficiency through the use of a variable area nozzle, whereas the axial machine would require a bypass of fluid and a reduced recovery for off-design conditions. However, a multistage machine would be more flexible with respect to design speed. Lower speeds can be used to match certain types of power absorbers without going through a gearbox, which is an important aspect from a cost and reliability consideration.

Thus, there are essentially two basic types of machines for this application, a high speed (variable area nozzle) radial machine and a low speed multistage axial machine, which is suitable for a direct drive of available power absorbers. The design problems of each are of the same order of magnitude as the M-1 pump design, where the high concentration of power involves severe stress and thrust balancing problems in addition to the effects of cryogenic temperatures on bearings, seals, and clearances.

1. Radial Turbine Design

High turbine efficiencies can be obtained in a single-stage radial machine, provided that the specific speed is a maximum practical value. As shown in Figure 15, efficiency increases with speed and decreasing diameter until about 25,000 or 30,000 rpm, but the gain is small above 20,000 rpm. Therefore, 20,000 rpm was selected as a design point for this part of the study.

From the standpoint of current power absorber designs, a much lower speed would be desirable in order to obtain a direct drive assembly. But this speed limitation (of 6000 to 8000 rpm) would result in a considerably lower efficiency machine, as indicated by the specific speed data. Therefore, the design of a radial machine was not considered to be restricted by power absorber speeds, and reduction gears would be used, if necessary. Thus, the problem statement which was

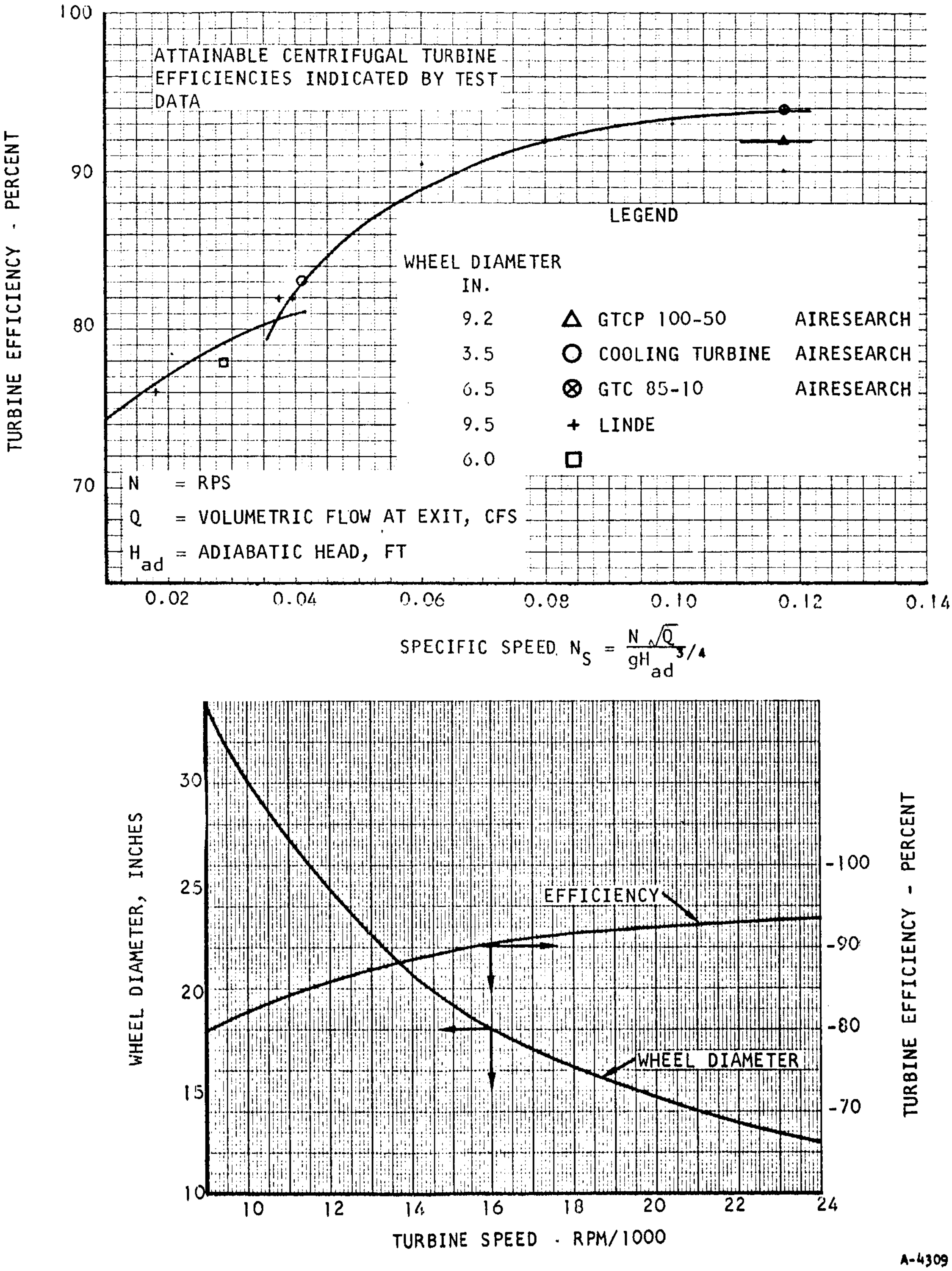

Figure 15. Centrifugal, Single-Stage, Turbine Design

assumed for the radial machine design is as follows:

Inlet Pressure, psia	1750
Discharge Pressure, psia	180
Inlet Temperature, °R	60
Flow Rate, lb/sec	588
Speed, rpm	20,000
Efficiency, percent	92

A preliminary design sizing for the radial inflow turbine was conducted, and a summary of the pertinent dimensions and fluid state points for the resulting 15-in. diameter impeller are given in Appendix C.

Figure 16 shows the corresponding state points on the enthalpy-entropy diagram.

The Reynolds Number based upon the hydraulic radius at the exit is

$$Re_{exit} = 2.7 \times 10^8$$

At these values, the roughness factor of the channel passages has a large effect on frictional losses. Therefore, very smooth or polished surfaces will be required.

As shown in Figure 17, the wheel is double-shrouded to reduce leakage flows and to minimize the thrust forces on the bearings. The labyrinth seals are designed for high pressure differentials with wear-in fins to minimize the final clearances.

The nozzles will be the variable vane angle type to provide for the range of flows and pressures produced by the test pump. Some losses will occur due to mismatches in velocity vectors at the off-design points but, in general, the overall recovery will be higher than that of a fixed area machine, since no bypassing of fluid will be necessary. A maximum amount of work will be extracted from all the fluid passing through the machine.

Performance

The predicted performance for the radial machine, equipped with variable area nozzle vanes and matched to the M-1 turbopump output, is tabulated in Appendix C.

It is to be noted that the variable area feature permits excellent recovery gains for all areas of the turbopump map. For the assumed basic turbine efficiency of 0.92, the total hydrogen recovery exceeds a minimum 75 percent of original inventory for any operating point.

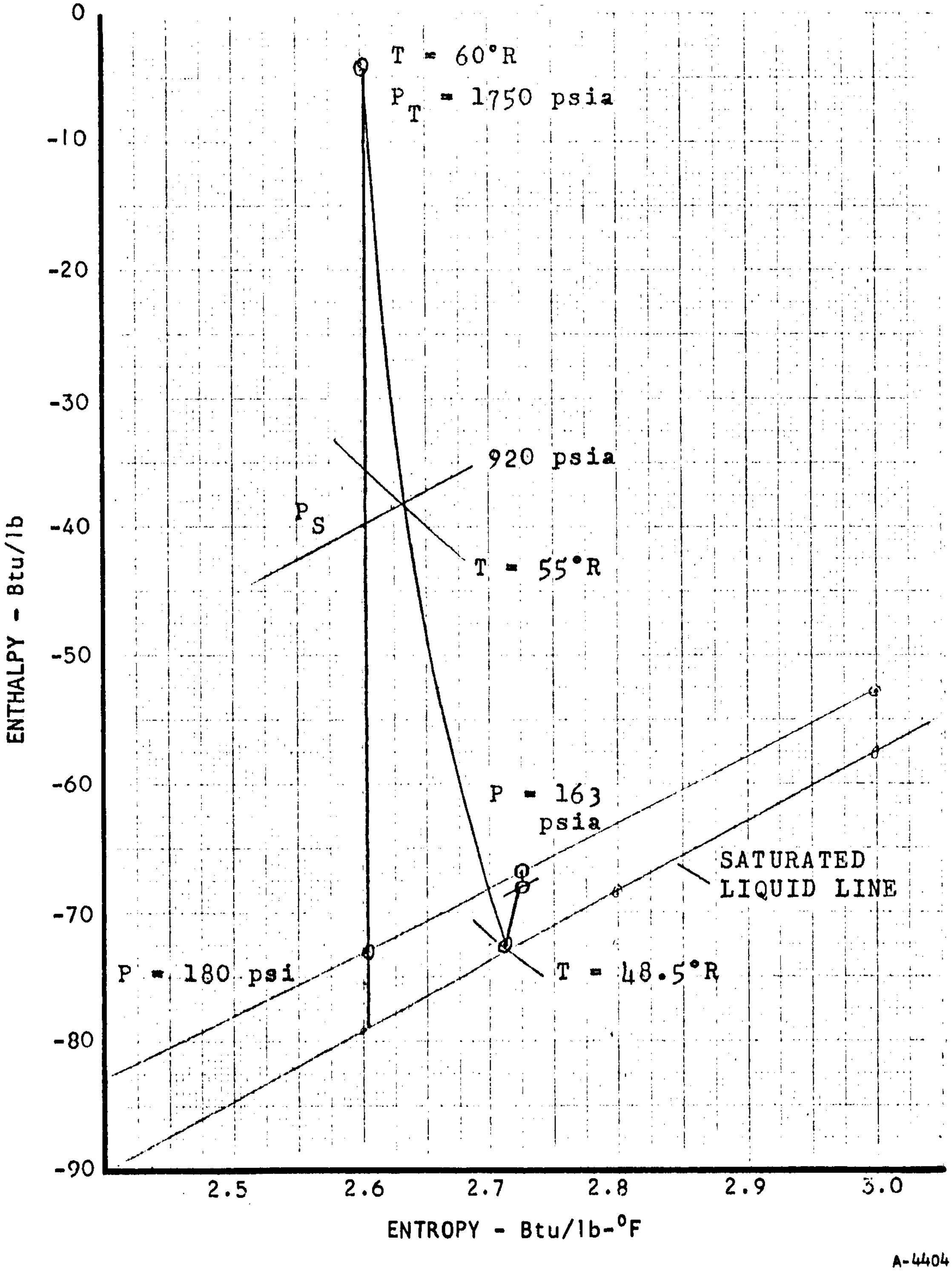

Figure 16. Radial Turbine Design

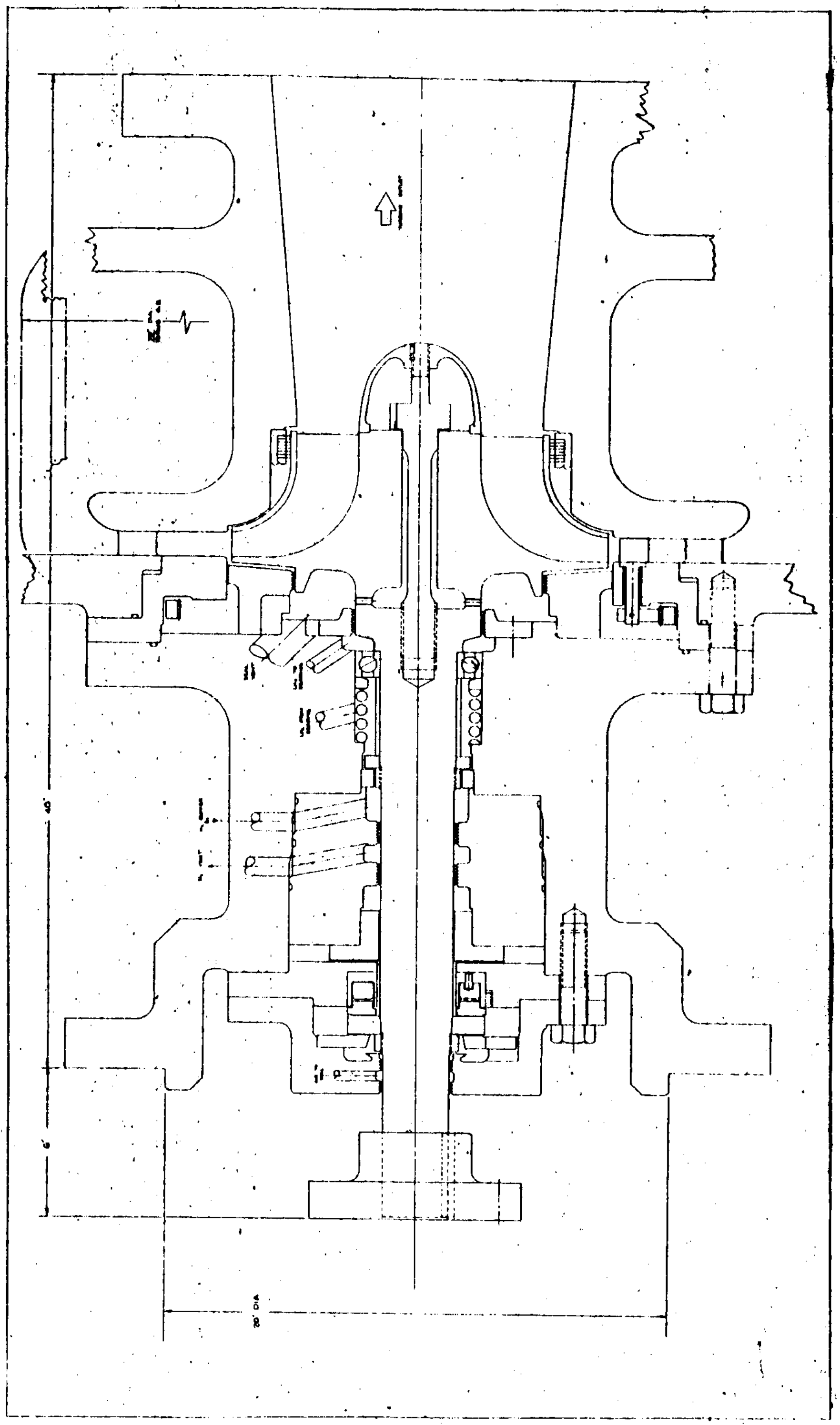

Figure 17. Radial Turbine Design

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Mechanical Design

The turbine wheel shown in the drawing is made of aluminum (Type 6061) and is fully shrouded to minimize leakage and to provide thrust balancing forces. The inlet blade angle is 90° and the inducer blade angle at the average discharge diameter is 40.5°. These are preliminary values and are subject to detail design modifications.

Thrust balance of the turbine is achieved by appropriate selection of the labyrinth seal locations on the inducer shroud and backside of the turbine wheel. The pressure drop across these seals is approximately 800 to 1000 psi.

The turbine wheel is mounted to an Inconel 718 shaft and drives through a gear type (Curvic) coupling. The shaft wheel attaching bolt is fabricated from Invar. The turbine shaft is supported in a two bearing system, one bearing is hydrogen lubricated and the other is warm oil lubricated.

The hydrogen lubricated bearing, at the turbine end, is a ball bearing which is preloaded thrust-wise by means of a spring and sleeve arrangement, as shown in the drawing. The bearing utilizes an Armalon separator, which has proved successful in various other cryogenic bearing applications. The output end of the shaft is supported in a sleeve type journal bearing. In addition, a "Kingsbury" type thrust bearing is incorporated to accept the transient unbalances which are incurred in start-up or shutdown as well as any residual thrusts that may be encountered from the loading equipment.

The sleeve and thrust bearings are lubricated by an external pressure oil system. The bearing cavities are separated by redundant seals. A carbon face bellows seal is used as the primary seal. The back-up seal is of a pressurized labyrinth design. In order to prevent contamination of either the oil or hydrogen which is used as bearing lubricants gas buffering of the bearing cavities is utilized by using pressurized hydrogen. The buffer gas is injected in such a manner as to provide controlled leakage in one direction only.

The turbine rotating group is housed in a structure fabricated from CRES 304 stainless steel. The turbine torus is a constant area channel which is internally stiffened with a series of guide vanes. The turbine nozzle is of the variable area type. The design is such that continuous modulation of the nozzle area is possible. It is intended at this time to preset and lock the nozzles during each test run. Redundant static seals are provided. These would include pressure actuated devices as well as conventional seal members. The turbine is equipped with a rigid coupling for direct connection with the driven equipment.

2. Axial Flow Turbine

In order to obtain a low speed machine that can be directly coupled to a low speed power absorber, an axial flow design was investigated. Multistaging permits lower blade velocities and rotating speeds, and a split flow design eliminates thrust loads and high pressure seal problems. Therefore, this approach was considered attractive if the recovery characteristics were acceptable.

Studies indicate a double (or split) flow machine with seven stages per side as the optimum configuration. The flow enters the turbine at the center from an inlet scroll which is connected to the 12-inch supply line. The flow splits, passes through the two sets of turbine blades, and discharges through exit collector plenums at both ends of the machine. The flow paths are then merged prior to entry into the 18-inch discharge line. This unit is shown in Figure 18. The tentative design speed has been selected as 6750 rpm. Based on state of the art design techniques, a total-to-total blading efficiency of 0.90 can be attained with this machine.

The unit is designed to drive two pneumatic or hydrodynamic loading devices which are mounted on either side of the turbine.

Detail Design of Turbine

Initial calculations indicated that if an adequate stage specific speed, N_S, is selected, a high value of blading efficiency could be obtained through the use of current blading design techniques. As a result of this, a six-stage axial flow turbine was investigated.

Several disadvantages were immediately apparent. One major disadvantage was the very large axial thrust forces generated by the turbine which would reach levels of ten to twenty tons depending on the particular match point between the test pump and turbine. This thrust could be decreased somewhat by designing constant pressure (or "impulse") hub sections for the rotor blading, but this is at the expense of lower attainable stage efficiencies. A thrust-balancing piston could be employed at the expense of increased shaft and bearing complexity, but the disc friction power required to drive this device would merely be fed back into the system as a heat input with a subsequent decrease in hydrogen recovery.

In addition to the above, the high pressure shaft seal is a considerable mechanical problem. As will be seen in a later section, the requirements for such a seal, operating under as much as a 2500-psi differential head, fall far outside the current state of the art in seal design.

Since the turboexpander system ultimately proposed to NASA must be of a "minimal risk" nature, these mechanical drawbacks, which are inherent in a single-flow axial concept (and to a lesser degree in the radial inflow turbine) must be minimized.

As a result of these considerations, the double-flow axial turbine is presented as an optimum turbine configuration. As can be seen from Figure 18, no high pressure seal is required and the axial thrust forces, as developed by each half of the turbine, effectively cancel one another at all operating points on the pump map. It is to be noted that as long as proper stage specific speeds are maintained for the double-flow concept, no significant decrease in efficiency below the level of the single-flow multistage axial turbine will occur.

The currently attainable performance of well designed axial flow turbines is given in Figure 19 in terms of stage total-to-total efficiency and specific speed. It is evident from this curve that to obtain the highest possible design efficiency, while at the same time minimizing the number of stages, the stage specific speed should not be lower than about 75. Thus, for a given rotating speed, N, and volume flow, Q_2, the number of stages required is readily determined and the speed of the axial turbine can be selected so as to properly match an available loader. The number of stages may then be adjusted to obtain an optimum stage specific speed.

Basing the decision on a survey of low speed loader characteristics, a speed of 6750 rpm was selected as the design speed of the turbine. With this speed, and the volume flows derived from the design point exit pressure of 180 psia, seven stages were obtained as optimum for the design head. Thus, seven stages per side must be employed for the double-flow machine.

Having determined the work required per stage, the selection of optimum blade geometry must then be made. This selection also has a direct bearing on turbine diameter inasmuch as changing the blading work coefficient from one of pure impulse to one of pure reaction requires a diameter increase of more than 40 percent. Generally speaking, the higher the reaction, the higher the efficiency. For a given flow rate, however, the resulting higher diameters will produce increasingly smaller passage heights with attendant increases in secondary and leakage losses. An optimum value of stage reaction, therefore, is implied by the foregoing trends.

In order to clarify the above relationships, an analysis was conducted with the goal of determing the best vector diagrams for the axial turbine staging. Details of this analysis are given in Appendix C.

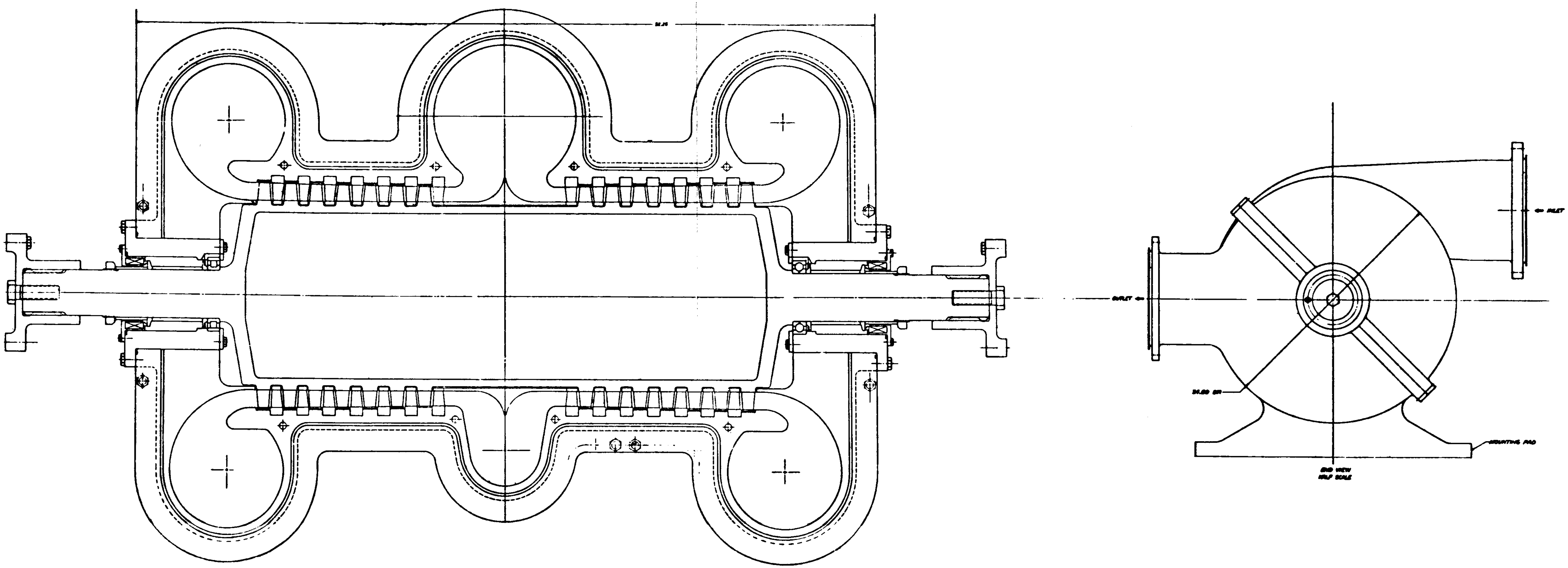


Figure 18. Multistage Double-Flow Axial Turbine, LH_2 Recovery System

SCALE 7:1 (APPROX.)

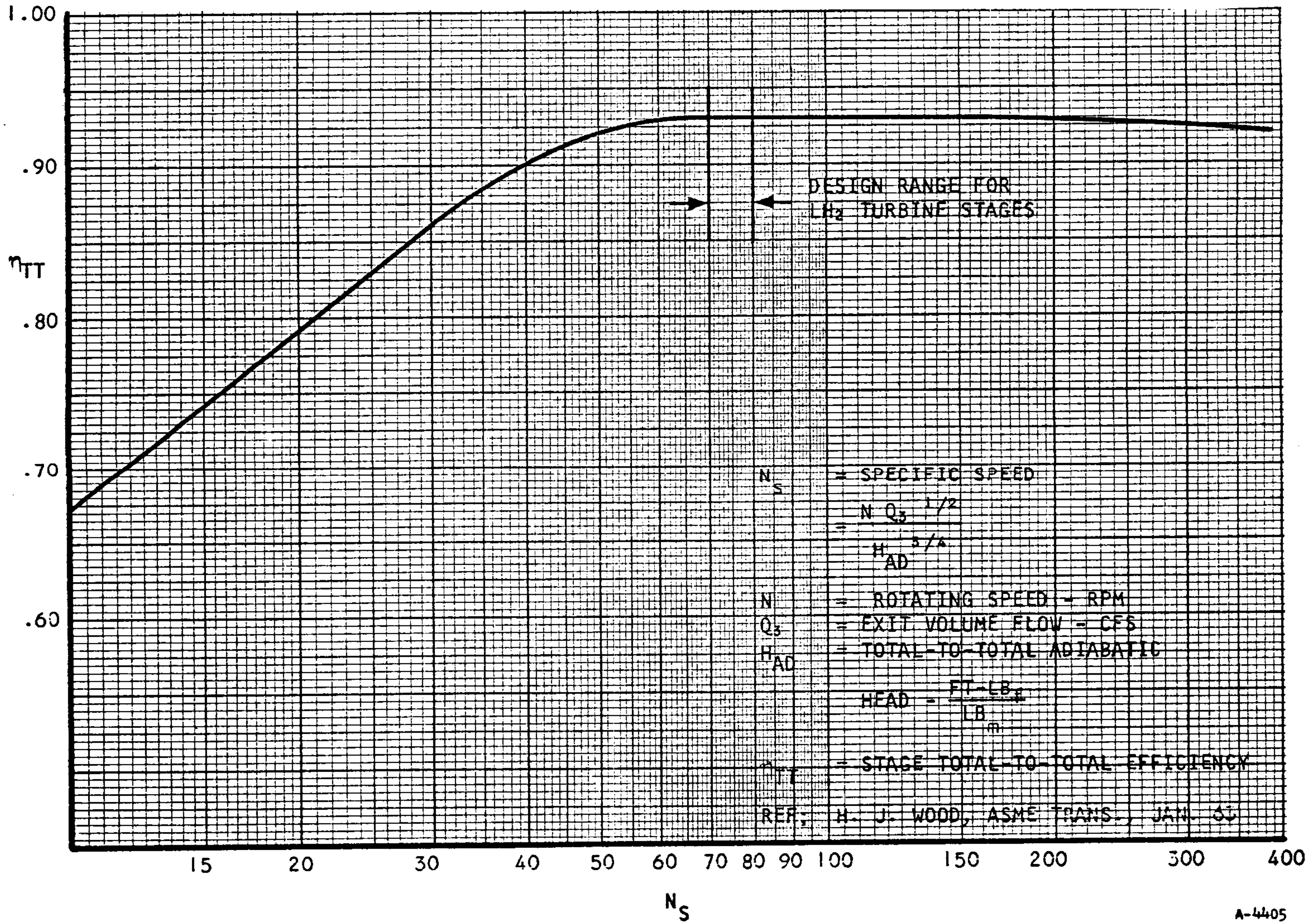


Figure 19. Maximum Attainable State-of-Art Performance for Single-Stage Axial Flow Turbines

The resulting optimum vector diagrams were calculated for a typical stage (in this case, the last) at the hub, mean and tip stations for values of $\alpha_{1h} = 70^{\circ}$ and $\lambda_h = 0.7$. These are shown in Figure 20 together with the pertinent diameters. The hub diameter which results is 13.34 in., and the tip diameter is 15.74 in. The corresponding tangential tip speed for the blades is 465 feet per second.

In order to greatly simplify the mechanical design, a constant hub diameter has been specified together with the same rotor and stator blades for all stages. Since the working fluid is virtually incompressible, very little change in annulus height is required.

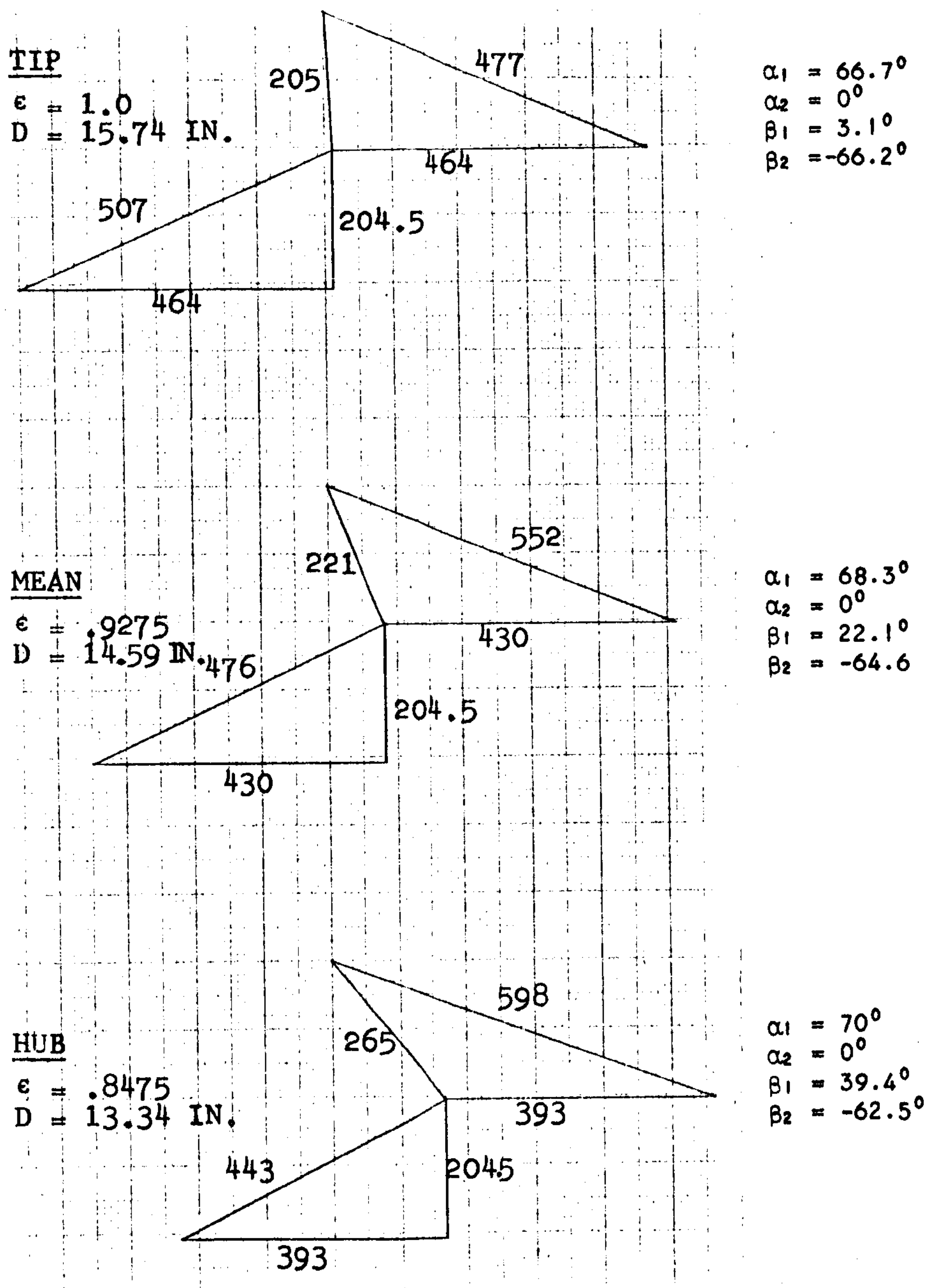
Using the optimum blade row solidities of Reference 3, it has been determined that ninety-seven (97) rotor blades are needed per stage for a chord length of one inch. In addition, eighty-one (81) stator blades are needed for a chord length which varies from 1.00 inch at the hub to 1.25 inches at the tip.

The first stage nozzles are an exception, however, as they must be matched to the flow velocity and direction emanating from the inlet volute scroll. Very little turning is required from these vanes, and because of this, uncambered profiles will probably be specified for the final design.

The recovery turbine is peculiar in that any energy available at the shaft must merely be dissipated in a convenient load. Because of this, such quantities as disc friction and bearing and seal losses, which are normally considered in a typical turbine design, are not pertinent factors in the present analysis. As seen in foregoing sections, improvements in hydrogen recovery over and above simple iso-energetic throttling are directly proportional to turbine efficiency, which means that a maximum "blading" efficiency is desired. Since the effect of reheat is very small, if not negligible, the stage total efficiency is a good approximation of the overall total-to-total efficiency. The estimated overall η_{T-T} for the double flow multistage axial turbine is thus 0.90.

Maximum "blading" efficiency is achieved by optimizing the vector diagrams and minimizing flow diffusion on the blade surfaces. In addition, a careful attention to the leakage over the blade tips is required. In a multistage axial turbine, leakage paths occur across both the stator and the rotor. In a typical turbine design, these leakage flows are controlled by multi-gland labyrinth seals for the stators, but in the rotors they must be controlled by minimizing the running clearance between the blade tip and the stationary shroud. The latter is more difficult and becomes quite critical for low blade heights. For the subject turbine, however, the very low stress levels in the rotor permit the attachment

α_1 = STATOR EXIT ANGLE
β_1 = ROTOR INLET ANGLE
β_2 = ROTOR EXIT ANGLE
α_2 = STATOR INLET ANGLE TO FOLLOWING STAGE



NOTE: VELOCITIES GIVEN IN FPS

A-4406

Figure 20. Design Vector Diagrams
Last Stage (VII)
LH_2 Turbine

of a rotor tip shroud band. The use of this ring and a mating labyrinth gland will give good control of the rotor leakage flows.

Detail analysis of the seals results in the specification of a six gland labyrinth for both blade rows (the same for all stages). Operation of these seals at a running clearance of 0.010 inch will limit the flow leakages to about one half of one percent of the annulus flow. Thus, the efficiency penalty per stage is limited to about one percent.

It is a well-known fact that the frictional losses in the blade passage of a typical gas turbine are directly affected by the level of the flow Reynold's number. The very low viscosity of liquid hydrogen results in extremely high values of Reynold's number, in fact so high that the frictional losses are independent of Re and are a function of surface roughness only.

The Re level of the rotor blades, based on the one-inch chord line is 5×10^7. For a minimum loss at this Re, a surface finish of 4 rms is required. Since it is unlikely that this level of finish can be achieved, it becomes readily apparent that the surface finish of the blades will have a direct effect on the maximum attainable blading efficiency. Surface finishes of 32 rms and 63 rms result in efficiency decreases of five and six points, respectively, below the theoretical optimum. The method of manufacture must be carefully considered in light of the above. Since all rotor blades are the same cast, one piece rotors of 347 Stainless Steel are attractive from the cost standpoint. However, if the as-cast surface finish of these parts cannot surpass 63 rms, then the rotors will probably be machined from Inconel 718 pancake forgings with high surface polish as a goal.

Cavitation Considerations

In the case of hydraulic turbomachinery, cavitation effects must be considered as a matter of course. This is particularly true in pump design, but not necessarily in the case of a hydraulic turbine where it is usually neglected. The reason for this is apparently due to the fact that cavitation effects are normally confined to regions near the saturation line which, for the turbine, means in the last stage, where the chances for subsequent damage are small. In addition, liquid hydrogen pump experience indicates that the damage resulting from cavitation of liquid gas is much lower than that noted in water pumps.

If the cavitation region is large enough, the liquid flow will be displaced and must then readjust to satisfy flow continuity. The resulting shift in design velocity distribution will result in a performance inefficiency.

A cavitation analysis was conducted for this turbine and some interesting comparisons were obtained. The details of the analysis are also given in Appendix C. The low volume flows and the low rotating speed indicates that cavitation will not be experienced even with a surface diffusion of 25 percent. Such is not the case for the radial inflow turbine, which implies that incipient cavitation may exist at the exducer shroud for design speed operation.

The data of Figure C-4 of Appendix C is of additional usefulness when considering decreases in turbine back pressure as a result of matching the pump flows in the high region of operation. Iterative calculations show that the exit total pressure can be reduced to 100 psi before incipient cavitation is in evidence on the blades surfaces of the last stage rotor.

Mechanical Design

Rotating Group

The turbine rotor is composed of fourteen rotor blade rings welded together by the "Electron Beam" process with a center separating ring, two end cones, and two output shafts (see Figure 18). Each rotor ring, the center ring and the end cones are to be case 347 Stainless Steel. A finish machining operation will provide the blade height and an adequate surface finish for each stage. The cast 347 provides good material and weld properties at cryogenic temperatures (-400°F) and also very good surface finish in the as-cast condition. The solid output shafts are forged Inconel 718.

A stress analysis on the rotor group shows the maximum radial blade stress to be 13,200 psi and a maximum tangential stress in the last stage rotor hub, due to blade and pressure loading, to be 42,700 psi, which is very conservative. The margin of safety is 40 percent of the material yield strength. This is for the worst possible pressure condition. Buckling and torsional analyses were also performed in the preliminary design of the rotor.

A critical speed analysis was performed on the rotor which indicated a first critical at 12,700 rpm, and the second at 17,700 rpm. Figure 21 shows the resulting bearing loads. It is to be noted that these speeds are well above the maximum possible turbine steady-state speed of 7820 rpm and thus will present no problem.

With the aid of a digital computer, a detailed stress analysis was made of the rotor with a goal of determining the tangential and radial stress distribution in the rotor end sections. The results of these calculations are shown in Figure 22 and 23.

The minimum hub burst speed for the turbine rotor end cones was found to be 42,800 rpm at -400°F. The last stage rotor rings have the lowest burst speed, which was found to be 21,000 rpm at -400°F.

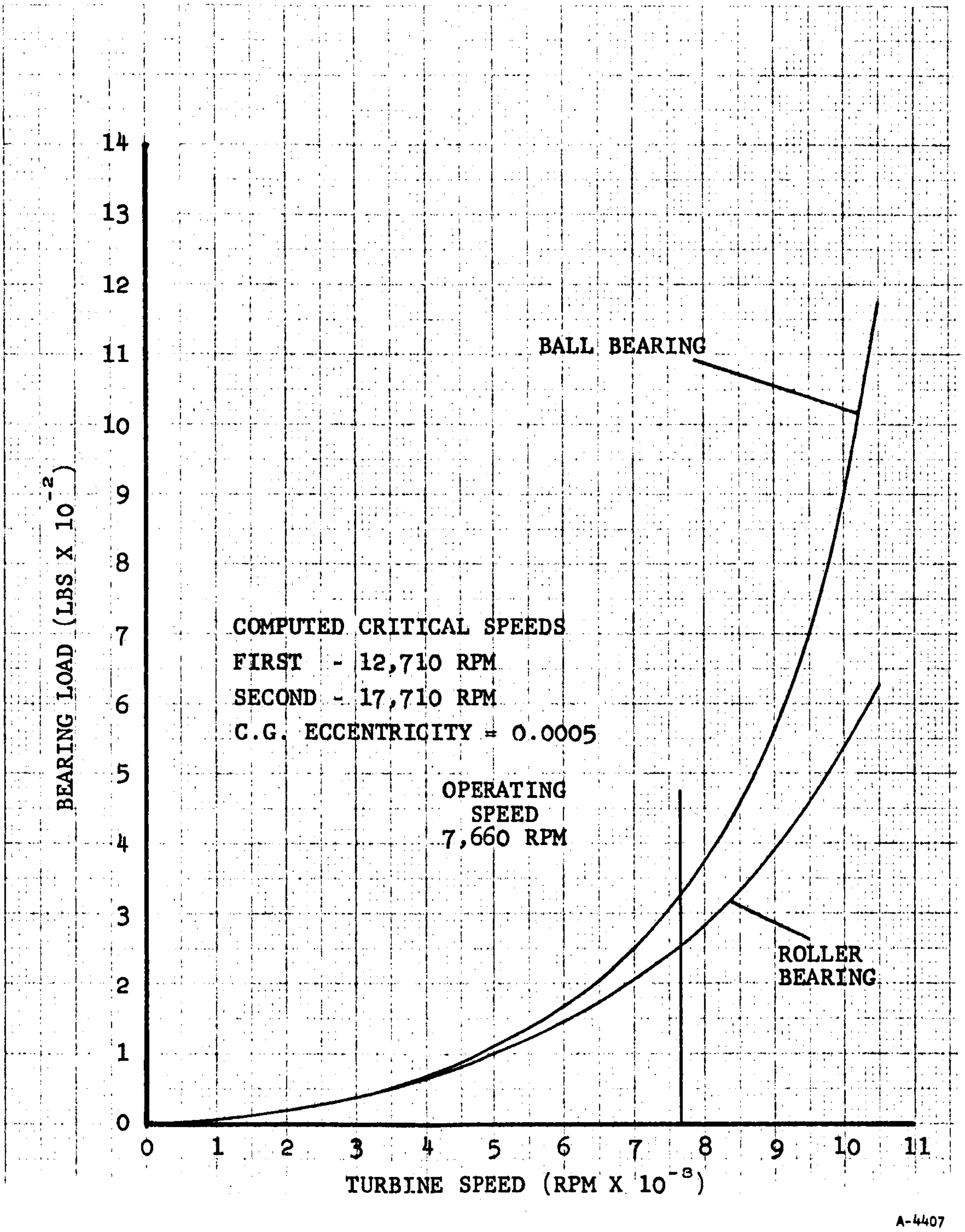


Figure 21. Cryogenic Turbine Bearing Loads

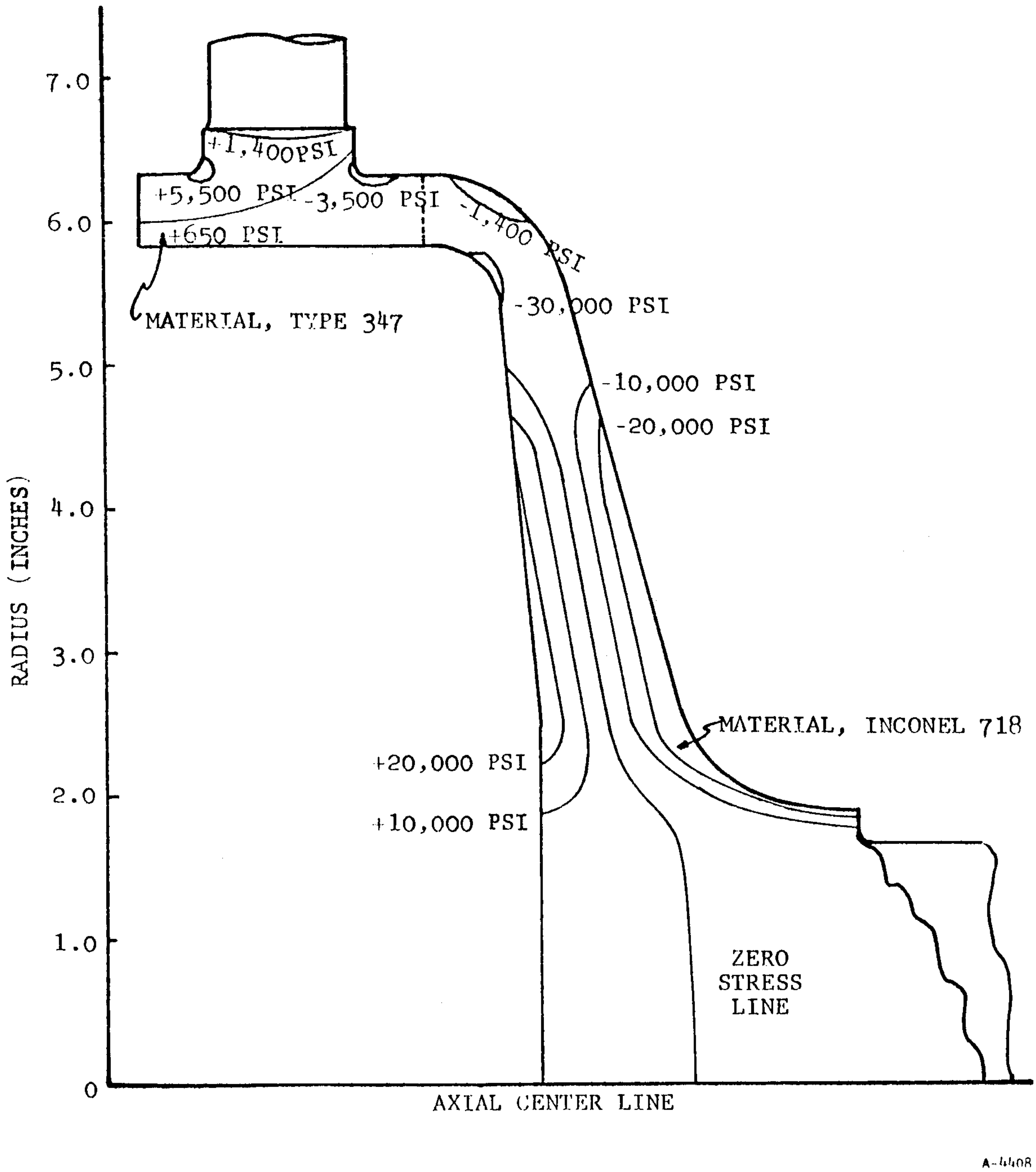


Figure 22. Radial Stresses in Cryogenic Turbine End Cone

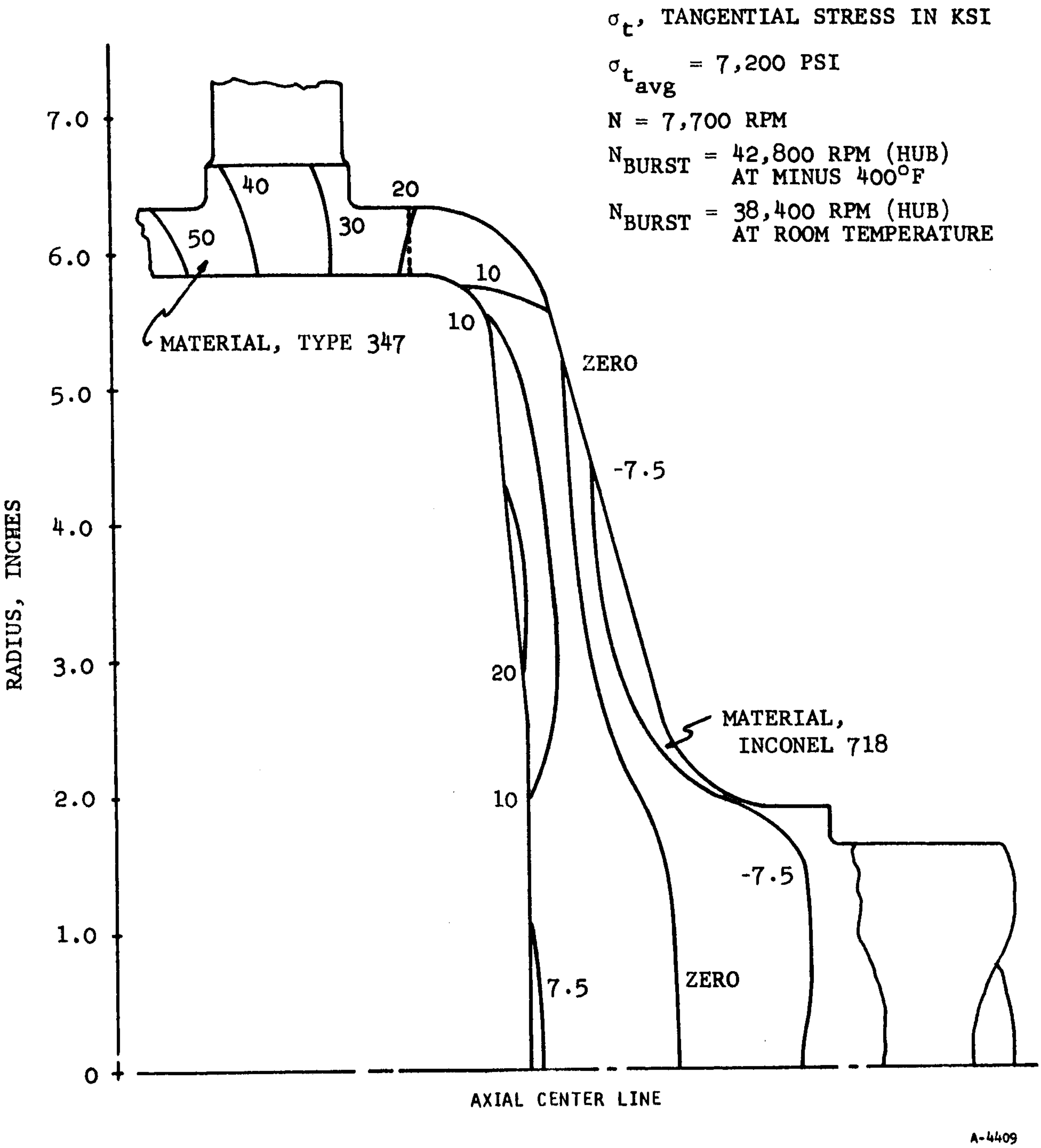


Figure 23. Tangential Stress Distribution in the Cryogenic Turbine End Cones

Again, it is to be noted that these speeds are well above the predicted turbine free-run speed, which can be conservatively estimated at twice the design speed.

The above data is impressive evidence of the anticipated high reliability and safety level of the multistage axial turbine design as compared to that of the radial inflow concept.

Turbine Housing

The turbine housing is a two-piece casting, made from 347 Stainless Steel. A continuous LH_2 seal is provided at the bolted parting surface.

Stator to rotor floating ring labyrinth seals are installed in machined slots in the turbine housing. These cast 347 stainless steel seals will accommodate any turbine housing warpage. Cast 347 stainless steel split ring stators are also installed in machined slots in the turbine housing.

A two-piece housing was selected to eliminate multiple seals and to facilitate turbine assembly.

A preliminary structural analysis was made on the housing to check the case deflections to be sure that running clearances were not excessive. A detail analysis on the inlet volute scroll will have to be made in the final design. Mounting pads are provided in the lower half of the turbine housing.

Bearings

The turbine will be supported by a ball bearing and a roller bearing, both of which will be lubricated by the liquid hydrogen working fluid at 180 psia. The ball bearing will take thrust loads in either direction and will axially locate the roller. Stack-up, deflection and thermal growths will be accommodated by the roller bearing, which acts as a slip joint at the other end of the rotor.

The following table summarizes the bearing selections and their operating conditions:

Type and Size	BALL BEARING Split Inner Race Extra Light Series	ROLLER BEARING Cylindrical Roller Extra Light Series
Bore diameter, mm	85	85
Maximum speed, rpm	8200	8200
Maximum DN value x 10^{-6}	0.70	0.70
Maximum load - Radial - lb	250	250
Axial - lb	1000	
Material - Race and rolling element	440C	440C
Separator	Aluminum shrouded Rulon or Armalon (both glass fiber reinforced Teflon)	

Theoretically, the thrust forces (about 10 to 15 tons per side) of the two sets of staging will cancel one another out. Due to manufacturing tolerances, however, it is probable that a force unbalance will exist. A thrust calculation has been made which indicates a maximum possible thrust unbalance level of 500 to 900 pounds, safely below the design limit of the proposed ball bearing.

Typical state of the art design techniques for cryogenic bearings are indicated including open-race curvatures which will minimize heat generation due to ball spinning and loose diametral bearing clearances which will serve to improve lubricant circulation.

Available data on cryogenic bearings operating on LH_2 and LN_2 show successful operation for smaller bearings subjected to equivalent loads and operating at the same or higher DN values.

Shaft Seals

The choice of a double-flow turbine for this application eliminates the need for a high pressure shaft seal for which the problem statement (2500 psi to ambient at -400°F, 10,000 fpm rubbing velocity) is extremely severe, and which can be solved only with a high level of system complexity.

The double-flow turbine requires only one low pressure seal at each end of the turbine shaft. Under all operating conditions, the pressure differential across the seal will be less than -200 psi.

An off-the-shelf, bellows-type, axial mechanical seal can be utilized. Specifically, a completely balanced two-ply bellows of Inconel 718 will satisfy the given requirements. In operation, the spring force of the bellows provides the face loading of the carbon sealing element. To date, the most successful operation in liquid hydrogen has been experienced using carbon Grade P5N.

A seal of this type must have a very carefully balanced bellows-carbon assembly so as to prevent excessive wear. Consultation with vendors in this regard discloses that each seal is balanced individually to guarantee optimum performance.

The rubbing velocity of the prescribed face seal is between 9000 and 10,000 feet per minute, and together with a maximum operating ΔP of 200 psi and temperature of about minus $400^{\circ}F$, indicate that successful operation is assured.

To supplement the above sealing technique, a loose multigland labyrinth seal is scheduled for use in series with the bellows face seal.

Miscellaneous

Heavy duty Sier-Bath flexible gear couplings will be provided to connect the cryogenic turbine to the loading devices. A sliding hub type will be specified so as to adequately take up the differential shaft expansion between the cold turbine and the relatively hot loaders. The crowned gear teeth of these couplings are more than adequate to account for any resulting shaft misalignment while successfully transmitting the turbine power.

3. Optimum Turbine

The primary objective of the analysis is to "perform a study of the technical and economic aspects of various liquid hydrogen recovery systems in sufficient depth to determine the relative merits and disadvantages of the various systems."

The economic merit of a given system consists of its ability to perform efficiently over the full range of the anticipated pump operating schedule. In the case of the two turbine expanders under consideration, the radial inflow machine is better equipped to do this because of its variable nozzle geometry. It must be pointed out that while the axial turbine performs at high efficiency for all pump off-design operating points, some of the latter result in the turbine being unable to expand either the entire available head and/or the entire available pump flow. Performance of both the radial and the axial turbines at their design points will be nearly the same, however, due to judicious selection of their respective design specific speeds.

Since the initiation of the comparative turbine study, it has been determined from Aerojet engineering that about 90 percent of the contemplated pump testing will be at design point. Because of this, and the fact that turbine performance for both geometries are comparable at their design point, there is little justification for accepting the mechanical risks of the radial turbine, which will be enumerated below.

In addition to recommending the system with the best overall economic potential, the technical risks, which will be encountered when trying to achieve this goal, must not be overlooked. The proposed system must be of such a nature that, when installed, long "trouble free" operation can be expected. In order to achieve this confidence level, the system components must be designed so as to avoid as many "problem areas" as possible.

In this respect, the multistage axial double flow turbine concept has the following advantages:

(1) Bearings--The lower operating speeds of the axial turbine permit DN operation which is in better alignment with the cryogenic bearing design state of the art as it is now known.

(2) Thrust--The high thrusts, both transient and steady state, are cancelled completely by the use of a double-flow turbine. In addition, each component of the axial turbine-loader system is thrust balanced independently of all other components. These conditions hold for all match points on the pump map.

(3) Seals--The use of a double flow turbine has eliminated the need for a high pressure shaft seal, which would present a formidable challenge in terms of current design ability.

(4) Stress Levels--The stress levels in the axial machine are less than one half of those found in the single-stage radial turbine, permitting a greater freedom in materials selection and mechanical design, as well as paying additional dividends in system reliability and safety.

(5) Cavitation--The axial machine will not cavitate at any point for the design back pressure of 180 psia, or for that matter, down to back pressures of 100 psia. In the case of the single-stage radial turbine, cavitation will probably occur at the exducer tip for all operating points. If this condition is severe enough, the flow must redistribute itself, thus upsetting design velocity conditions and resulting in performance inefficiencies.

(6) Speed Reduction Gears--Since the axial turbine approach is a direct drive solution, no highly loaded speed reduction gears are necessary to match the low speeds of the air compressors or water brakes.

The following advantages may be listed for the radial machine:

(1) Inherently Simple Design--The radial wheel and blading is more rugged and less critical from a dimensional standpoint.

(2) Low Development Cost--The radial wheel is easier to build, and having fewer critical tolerances, will cost less than an axial machine. Blade angles and contours are less important than in an axial machine, which means a considerable lower sensitivity to fabrication methods and techniques.

(3) Wider Range of Efficient Operation--The wide range of efficient operation is an inherent characteristic of a radial flow machine, when equipped with variable area nozzles.

Since high system reliability and safety are of paramount importance in the successful operation of the proposed LH_2 recovery system, the above arguments leave little choice but to select the multistage axial machine as the optimum turbine for the system. Its high performance level and mechanical ruggedness form a satisfactory solution to the basic problem statement.

IV POWER ABSORPTION EQUIPMENT

The preceding sections have shown that the LH_2 recovery turbine will generate approximately 52,000 hp under normal operating conditions and can produce as high as 80,000 hp at off-design conditions. While the absorption of this energy requires the use of loading equipment which is not available on an "off the shelf" basis, no "technological breakthroughs" are foreseen in the design.

Three general classifications of machines have been considered as suitable for this application, namely: electric, hydrodynamic, and pneumatic. The foregoing are all low-speed devices and, hence, are well suited to the needs of the multistage axial turbine.

Electric Generators

The primary advantage of the eddy current brake or inductor generator is the relatively simple controls involved with its operation. It is necessary only to vary the excitation of the d-c field to change the torque characteristic of the dynamometer. The control is smooth and stepless over the entire horsepower range and is controllable to within 50 rpm at all operating points. Operational experience with these units in the field has found them practically maintenance-free because of their simplicity.

For the current application, however, two inductor units would be required, one operating at each end of the double flow turbine. For proper sizing, the generators would be required to operate at 1800 rpm, thus necessitating a step-down gear box with a high power transmittal capacity. In addition, the energy generated by the turbine is ultimately absorbed by water used to cool the load absorbers. Because of this requirement, the high complexity of a cooling system with a large water reservoir is needed.

The gearing and size of the generator rotors present a very high moment of inertia to the turbine and will undoubtedly result in relatively slow startup times.

Investigation of the cost of such a system indicates that it would cost two to three times that of a comparable hydrodynamic or pneumatic device.

Thus, it is seen that the disadvantages far outweigh the few advantages for the electric generator loader, and as a result, it has been rejected for the present application.

Hydrodynamic Loaders

The devices using water as a load dissipation medium can be generalized into high speed and low speed water pumps. A special adaptation of the latter is the common water brake.

The former machine received attention while the high speed radial inflow cryogenic turbine was being investigated; and since the latter has been discarded as a suitable expander, the high speed pump will not be considered here.

A large low speed water brake has been designed by Kahn and Company of Wethersfield, Connecticut. This brake is designed to absorb 30,000 hp at 6000 rpm, which makes it suitable for absorbing a portion of the output of the LH_2 turbine under direct-drive conditions. It can be used at speeds up to 8000 rpm, will operate in either direction of rotation, and has a control range ability of approximately 15 to 1 with a maximum torque capacity of 30,000 ft-lb. The possible performance spectrum of this brake is given in Figure 24.

The present application will require two water brakes operating, one at each end of the double flow turbine. As in the case of the electric generators, a sizable water supply system is required for any mode of operation. The details of the system will be enumerated later.

The initial cost of the water brake loading system, estimated at between $150,000 and $200,000, makes it very attractive and worthy of further consideration.

Pneumatic Loaders

As originally envisioned, the pneumatic loader, or air compressor, would assume the form of one of the commercially available single-speed multistage axial flow compressors such as those used in either the J-47 or J-79 jet engines. A preliminary study quickly showed that, while only two J-79 units were needed to absorb the design power of the LH_2 turbine, a total of four J-47 compressors would be required.

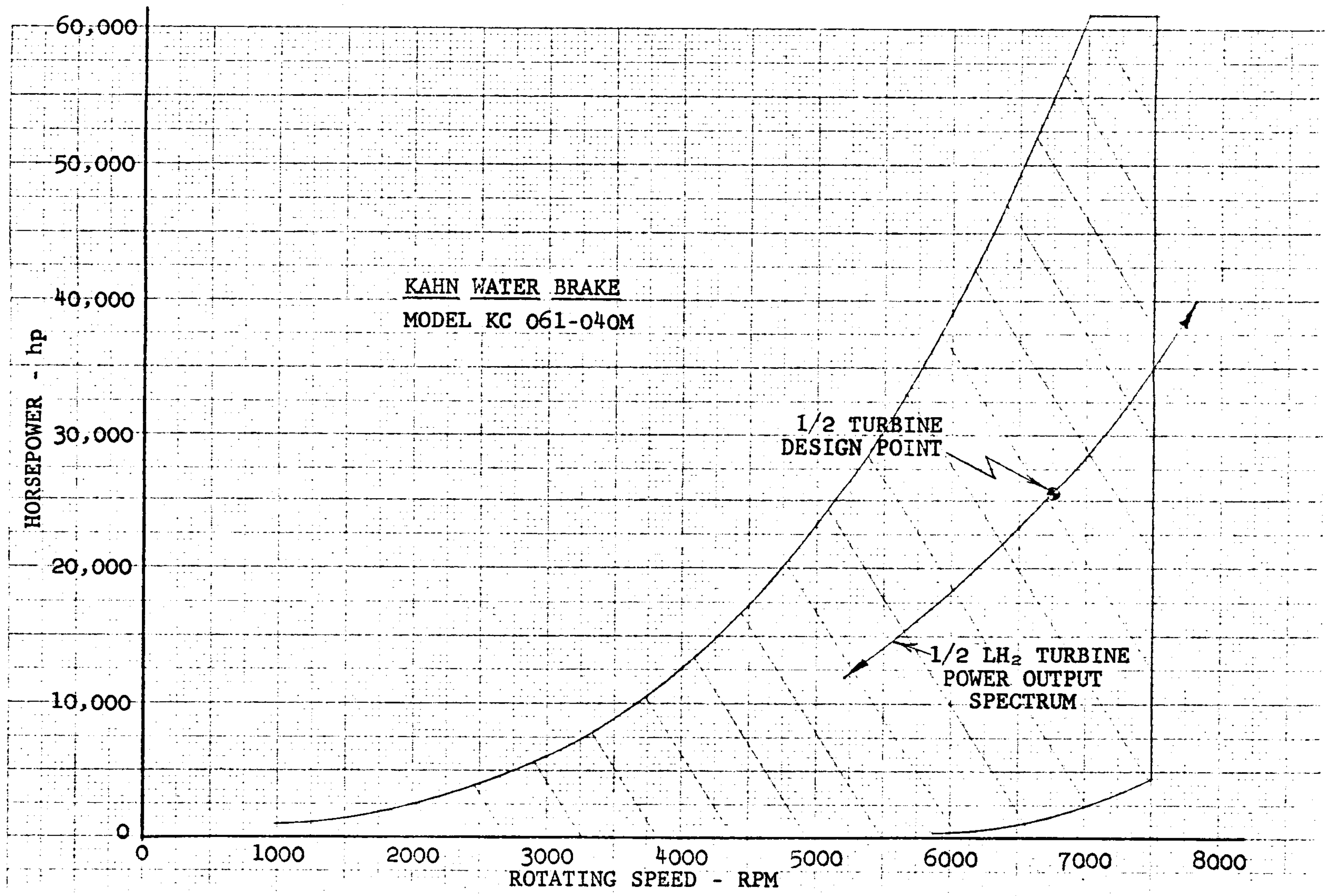


Figure 24. Water Brake Power Absorption Envelope

At this point, the study concentrated on the application of the General Electric J-79 compressor, which could be directly driven by the turbine. Consultations with GE personnel revealed a number of problem areas, the foremost of which appeared to be the limitation of the torque input flow path to the rear (aft) end of the compressor. To provide the proper rotation for the units (which are unidirectional as compared to the water brakes), 1:1 reversal gears must be employed for one of the two compressors required. An alternative to the gears would be a complete mechanical redesign of the inlet end of the compressor, which in the opinion of GE engineering would be very costly and time consuming. The gears would be roughly three feet in diameter and would result in pitch line velocities of over 50,000 fpm.

High thrust levels of up to 15 tons can be generated by the J-79 compressor and must be compensated for by the inclusion of specially designed heavy duty Kingsbury thrust bearings.

In order to guarantee surge-free starts in the jet engine, variable inlet guide vanes and six stages of variable stators are utilized, thus greatly increasing the controls complexity of this unit. These controls tend to impart a horsepower characteristic to the compressor which varies with about the fifth power of operating speed as compared with the cubic output of the hydrogen turbine. The effect of this is to greatly foreshorten the possible operating range of the J-79 unit.

Horsepower-speed lines of the J-79 are given in Figure 25 for various guide vane settings and a fixed downstream orifice. The design power variation of the LH_2 turbine is superimposed on these curves. The mismatch is apparent at the lower operating speeds. In order to better match in this range, i. e., without compressor surge, a higher design speed for the turbine is required. This, however, will result in a maximum turbine speed (at maximum pump head) that will exceed the tolerable overspeed of the J-79 compressor, which is currently rated at 7916 rpm.

These facts, coupled with the high cost (about three times that of the water brake system) of the modified compressors, tend to detract from its usefulness as a load absorber for the present system.

A pneumatic loader, nevertheless, remains very attractive for the LH_2 recovery system because of its very low installation and control complexity as compared to that of the hydrodynamic system. This

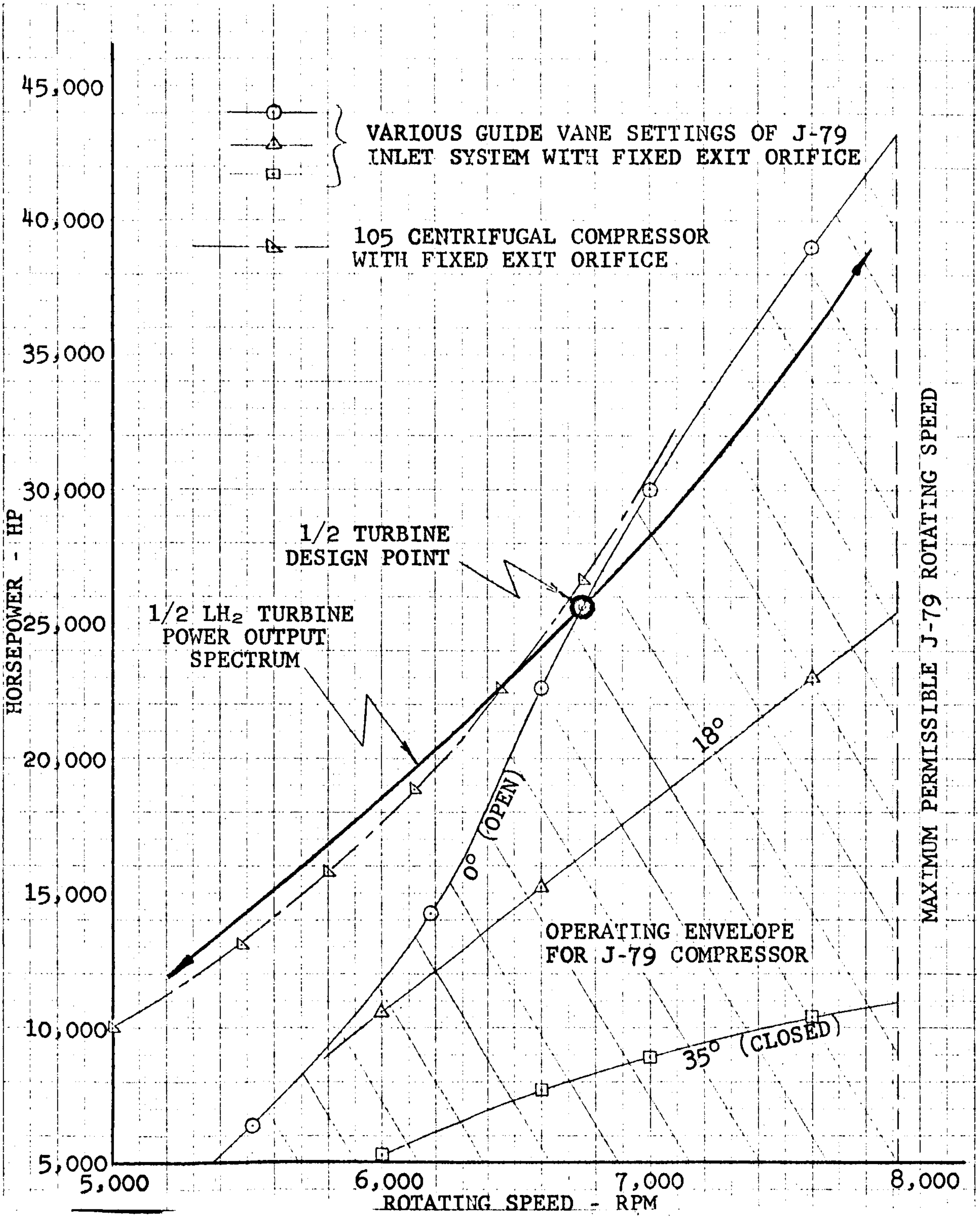


Figure 25. Various Loader Power-Speed Characteristics as Compared to LH_2 Turbine Output

is made possible by an abundance of the working fluid, namely air, available at the installation site.

As the J-79 unit became increasingly unsuited with regard to fulfilling the problem statement, a study was initiated which considered the application of a scaled-up model of one of AiResearch's many centrifugal gas turbine compressor rotors. The double-entry 105 compressor has been chosen as being best adapted to this system. The impeller has backward-curved vanes which tend to impart a wider operating range than a radially bladed unit delivering the same flow conditions. The compressor is a proven design and has been built in large production quantities.

To fit into the LH_2 system, the unit must be scaled up at constant tip speed by a factor of about 5:1, producing an impeller approximately four feet in diameter (similar to the J-47 compressor). The suitability of the matching of the scaled-up 105 impeller to the power requirements of the LH_2 system is indicated by the fact that, at 6750 rpm, the turbine power is absorbed at a corrected compressor speed (to design speed) of 1.04. As in the case of the J-79, two units will be needed to absorb the full turbine output. A layout of the proposed compressor is given in Figure 26.

Because of the dynamic similarity of the scaling, the stress levels of the scaled-up unit are the same as those experienced in the gas turbine application, which are minimal at best. No compressor failures have occured in this unit in over 70,000 operational hours or an accumulation of 600,000 engine starts. The foregoing figures are based on the experience gained from the operation of some 350 105-gas turbines in the Navy alone (the Army and the Air Force also use this engine).

Flex-mounted bearings with a spring rate of 50,000 pounds per inch are to be supplied with the compressor. They will result in a shaft critical speed of 1800 rpm which is well below the anticipated operating range.

The advantages of the 105 unit as compared to the J-79 compressor are many, and are summarized briefly as follows:

(1) Gears - No gears are required, either of the step-down or reversal type, as the unit can be conveniently driven, directly, from either end.

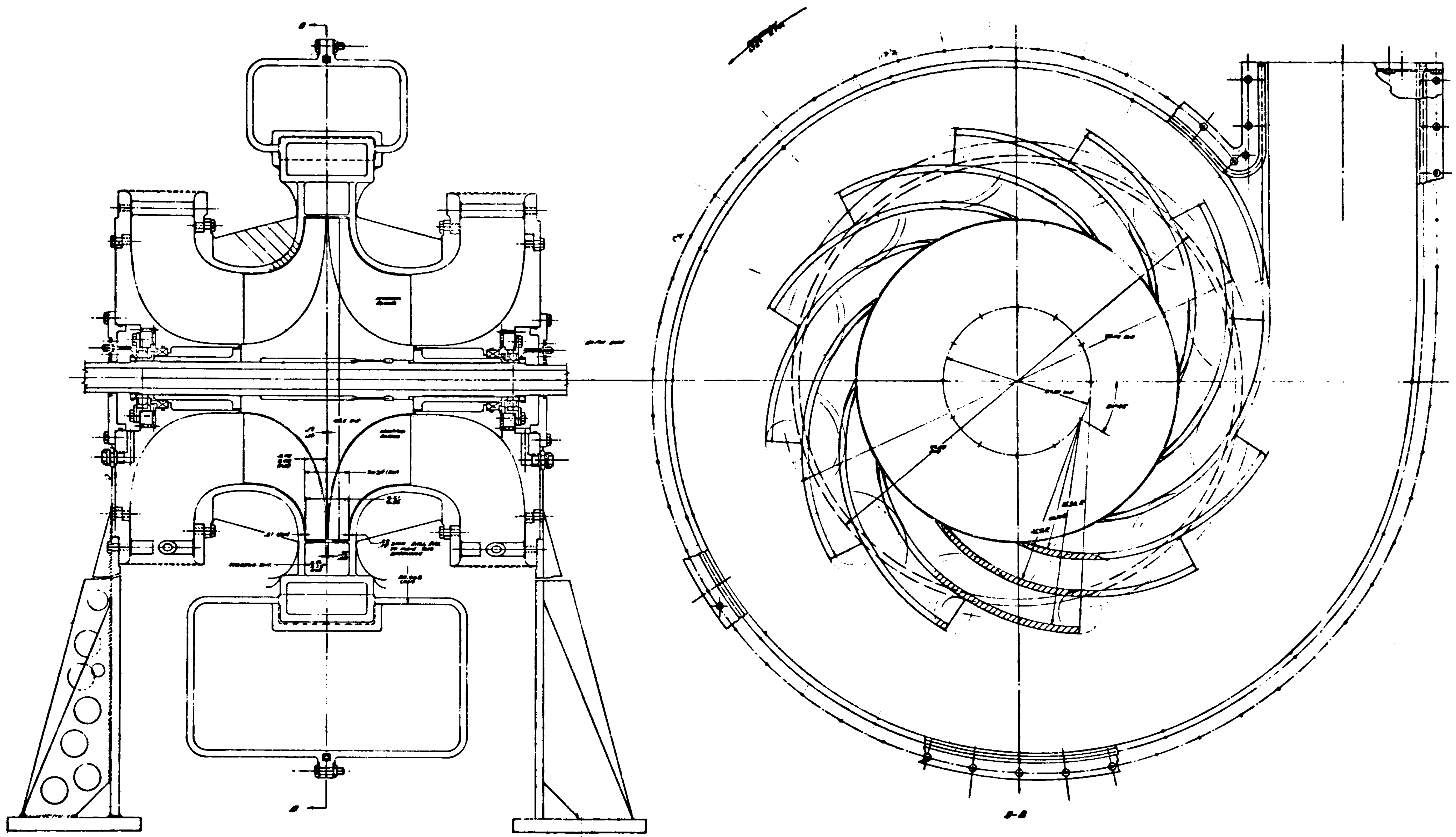


Figure 26. Double Entry 105 Centrifugal Air Compressor

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(2) Thrust - The high end thrusts normally generated by a compressor of this size are reduced to zero by the double-entry concept, in much the same manner as those of the axial turbine itself.

(3) Variable Geometry - The 105 centrifugal compressor develops a pressure ratio slightly in excess of 3:1, as compared to 12:1 in the J-79. The nature of the resulting surge characteristics, coupled with the wide range offered by the backward curved blades, eliminates the need for variable inlet geometry. The starts of the 105 unit will be surge-free.

It is of interest to compare the horsepower-speed characteristics of the scaled-up 105 impeller with power output of the turbine. The power characteristic for the 105 compressor, at an arbitrarily assumed exit orifice area, is shown in Figure 25 together with the turbine output power characteristics. It is readily seen that the match to the turbine is far superior to that of the J-79, and nearly follows a cubic relationship.

The above facts, coupled with the knowledge that there is a price differential (installed) of about three to one favoring the centrifugal unit, provide sufficient reason for the 105 unit to supplant the J-79 as an optimum pneumatic loading device.

The economy of the 105 design is enhanced by the fact that the large impeller can be machined using existing equipment. A further advantage of the scaled-up unit is that the tolerances can be easily adjusted for optimum procurement while not sacrificing the load-absorptive nature of the compressor.

Optimum Loader

The selection of an optimum loader among the many possible devices proved to be a difficult task in the sense that a more detailed study was required than originally anticipated. During a considerable period of the study, the J-79 compressor appeared to be the best power absorber and the study of other methods was discontinued. When the J-79 compressor modification was finally proved excessive, the water brake approach was hastily revived and the study of centrifugal air compressor was initiated. The subsequent discovery that a large water brake had been built for General Electric by the Kahn Company was encouraging and a detailed analysis of this approach was undertaken, as discussed previously. As a result parallel efforts were made on the two systems to determine the technical and economic feasibilities, and not until these studies were essentially completed was it possible to establish a preference for the air compressor system.

The optimum loader for the LH_2 recovery system is one which has the following characteristics:

(1) Smooth load management over entire spectrum of turbine-pump operating regime

(2) High reliability

(3) Low maintenance

(4) Low cost

Among the loaders under consideration, only the water brake and the centrifugal air compressor seem capable of adequately satisfying the first item, and are therefore worthy of further consideration. The choice of the optimum device rests on the comparison between the two loading concepts in regard to Items 2 through 4. To aid this comparison, a detailed system layout has been prepared for each approach. These layouts are shown in Figure 27, and include all the system components with the exception of controls circuitry and the various associated "black boxes."

On the left is the water brake system with a water supply that provides a flow rate of over 2000 gpm to each brake. The single tank provides cool water from the bottom of the tank and receives the hot discharge at the top where the water stratifies in the same manner as in conventional household hot water tanks. Between runs, the water cools for reuse. A small pump can be used to promote circulation between runs if necessary. In accord with the manufacturers recommendations, separate pumps and pressure control valves are used for each brake. (In the cost analysis, the water system for the water brakes is referred to as a "water loop.")

Since the reliability of any mechanism or system of components is generally inversely proportional to the number of parts involved,

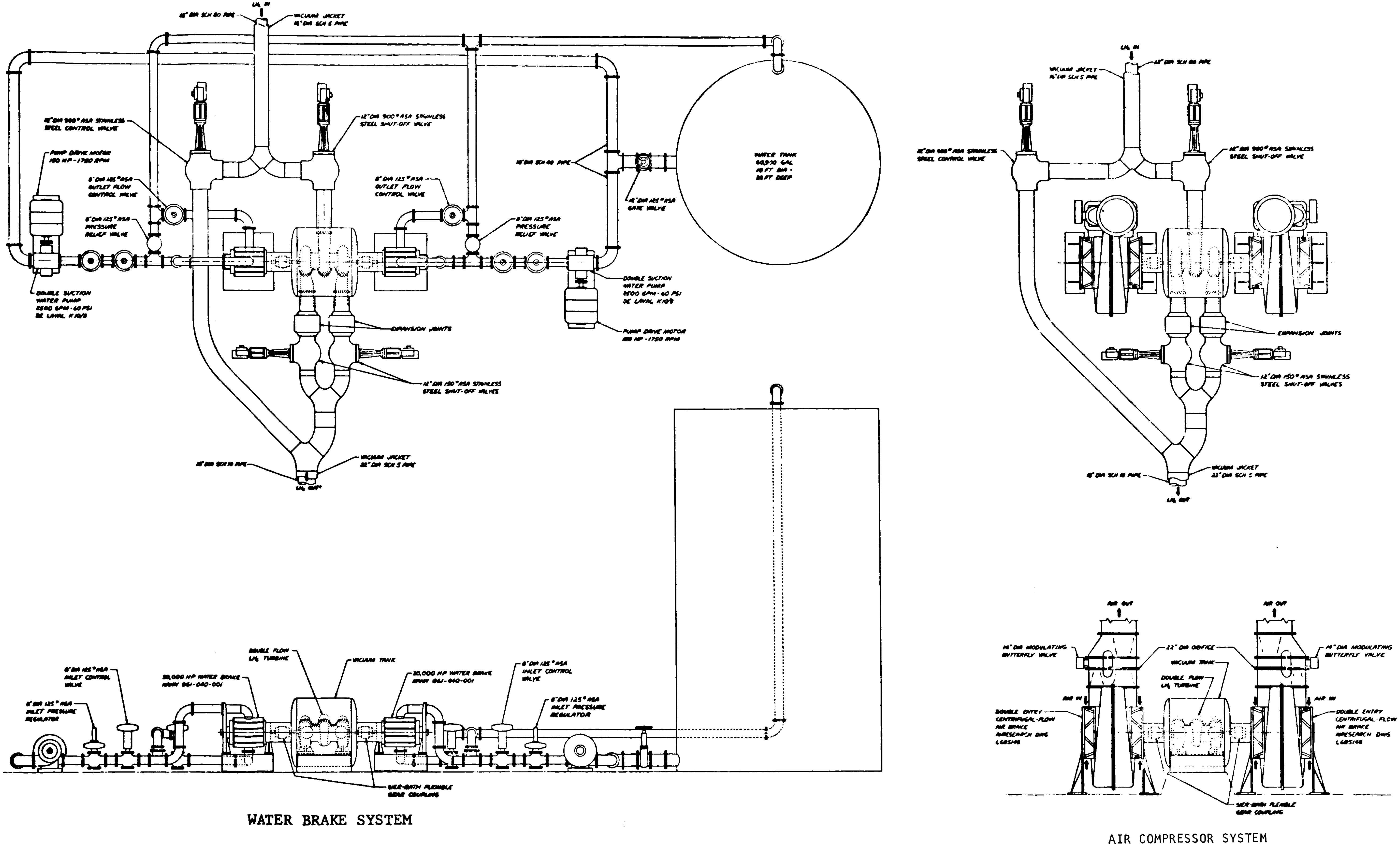

Figure 27. Comparison of Pneumatic and Hydrodynamic Loader Systems

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particularly when they are moving parts, the water brake system is clearly at a disadvantage. A comparison follows:

Water System	Air Compressor
1 60,000 gallon tank	2 control valves
1 12-in. gate valve	
2 water pumps	
2 150-hp electric motors	
2 pressure regulators	
4 control valves	
2 relief valves	

Contrary to popular opinion, the large water brake considered herein cannot be considered "off the shelf" in a fully developed sense. Many installation problems can arise.

Each water brake requires two remotely controlled flow control valves, one on the inlet and one on the outlet of the water brake. The control of these valves is very sensitive, particularly the back pressure valve. In the event of an overpressure (high shaft loading) condition, the brake casing must be protected by a bypass emergency relief valve, set to actuate at pressures above 350 psi.

Once the valves are calibrated, the brake performance is affected only by a change of inlet water pressure. In order to minimize the effect of pressure fluctuations in the water supply header, which can manifest themselves as severe vibrational forcing inputs to the rotor and hence possibly to the cryogenic turbine, a pressure regulator must be installed upstream of the inlet control valve so as to guarantee a steady inlet pressure of at least 30 psig.

Copius amounts of water are required to maintain temperatures of less than 180°F at the brake outlet. In the present system, the test site requirements include the need for a 60,000 gallon tank (18 feet in diameter and 32 feet high) and two electrically driven 2750 gpm pumps.

Water brakes are very sensitive to internal cavitation, which results either from an insufficient water supply, or poor internal design. The onset of cavitation can result in severe internal damage and rotor vibratory problems. The large unit currently under development at GE Evendale, for instance, is known to be experiencing problems of the latter type.

It is expected that the water brake system will require more maintenance than the pneumatic system. GE experience indicates that it takes two men almost full time to keep a large water brake system in operation. To quote GE personnel: "the decision to go this route (i. e., water brake) was on initial cost but, since installation, maintenance costs have long surpassed any initial saving."

As a result of the foregoing discussion, it is concluded that the water brake system will require more maintenance and will probably be less reliable in operation than the air compressor. Moreover, any slight advantage in initial cost of the former over the latter is likely to be quickly surpassed after a short period of operation.

Fail-safe operation of the centrifugal air compressor, of course, is a first order requirement of the LH_2 system. Catastrophic hub bursts of this impeller are very unlikely for this application. The operating tip speed of the 105 unit, being driven by the LH_2 turbine at its design point, is 1426 fps. Whirl pit bursts of the gas turbine impeller result in critical tip speeds of more than twice these values. At maximum turbine speed, 7820 rpm, the impeller speed is only 1651 fps. It is normal production procedure, for instance, to overspeed all 105 wheels to tip velocities of 2260 fps.

Failure modes can occur either of two ways:

(1) A loss of both couplings or shafts, or

(2) A loss of one coupling or shaft.

In the event of the first instance, the compressors will merely slow down to a halt. In the second case, the one remaining unit will rematch the turbine output at a lower torque level, but at a higher operating speed, about 1.24 times design speed. This means, that for the maximum pump head operating point, a failure of one shaft will result in a maximum possible loader tip speed of 2048 fps, well below the critical value.

It will also be impossible for the loaders to operate within their surge regions, due to the choice of the compressor control system which consists of a fixed orifice and a modulating control valve. This system will be described in detail elsewhere, but suffice it to say that in the event of loss of valve hydraulic pressure, the valve is so designed as to "fail open." Thus, a maximum exit control area is seen by the compressor, forcing it to operate in the "choke flow" region as opposed to the "surge" region. This procedure is used during the startup cycle with the valve then slowly closed until the operating point is reached.

In summary, it is clearly seen that from the standpoint of operational safety, the centrifugal air compressor has much to offer and is truly "fail-safe."

In view of the foregoing comparisons, the air compressor is selected as the optimum loader for the LH_2 recovery system.

V SYSTEMS ANALYSIS AND INSTALLATION

The original intent of this design study was to perform an analysis on a variety of systems in sufficient depth to determine an optimum system for the Aerojet test stand and then perform a detailed design study of the selected system. Various types of turbines and power absorbers were to be considered, including a combination of turbine and heat exchangers, using both new and currently available hardware.

As the study progressed it became evident that preliminary efforts were inadequate, and that several systems would require a more detailed analysis before a logical selection could be made. During this period the following were investigated:

(1) Use of Aerojet Lox Pump as a turbine

(2) Single stage radial turbine (high speed)

(3) Multistage radial turbine (low speed)

(4) Multistage axial turbine (low speed)

(5) Jet engine compressor power absorber

(6) Water pump power absorber (high speed)

(7) Water brake power absorber (high speed)

(8) Water brake power absorber (low speed)

(9) Gear boxes for combinations of high speed turbine and low speed power absorbers.

(10) Heat exchanger using liquid hydrogen (subcooler)

(11) Heat exchanger using tank boiloff vapor

(12) Combination heat exchanger and turbine, upstream and downstream locations.

At the midterm report period, these systems had been reduced to the following.

(1) Heat exchanger using tank boiloff vapor

(2) Axial turbine and two J-79 compressor power absorbers

(3) Radial turbine and two high speed water brakes

(4) Radial turbine with variable area nozzles, two gearboxes, and two J-79 compressors.

At the midterm period, NASA selected the axial machine with two J-79 compressors for detailed study and requested a continuance of the heat exchanger system design to the extent that funds would permit.

Further investigation of the J-79 compressor revealed problems which reduced the advantages of this approach, and a design study of a centrifugal air compressor was initiated. Also, the use of a low speed water brake was reconsidered, since this unit had been built and was available at a low cost.

Meanwhile, control studies indicated that there were two locations for the turbine, one at the present orifice downstream of the M-1 main control valve, and another location upstream of these control valves. These have been labeled in the following section as "Standard" and "ultimate" control systems, respectively. Each location has certain advantages and disadvantages and a clear choice is as yet undetermined at this time.

As a result, the following systems have been studied in sufficient detail to indicate technical and economic feasibility.

(1) Heat exchanger system

(2) Axial turbine with water brake absorbers

(3) Axial turbine with air compressor absorbers

(4) Turbine system with standard controls

(5) Turbine system with ultimate controls

Figure 28 shows a schematic diagram of these systems, and shows the present simple recovery system as a basis for comparison. From this diagram, the relative complexities as a result of the integration into the basic system are apparent.

An important aspect of design is the feasibility (and cost) of installation. The Aerojet test facility is essentially a completed structure and various modifications will be necessary for incorporation of any of the systems listed above.

The following sections present the basic preliminary designs for the installation of the heat exchanger and turbine system in the Aerojet facility.

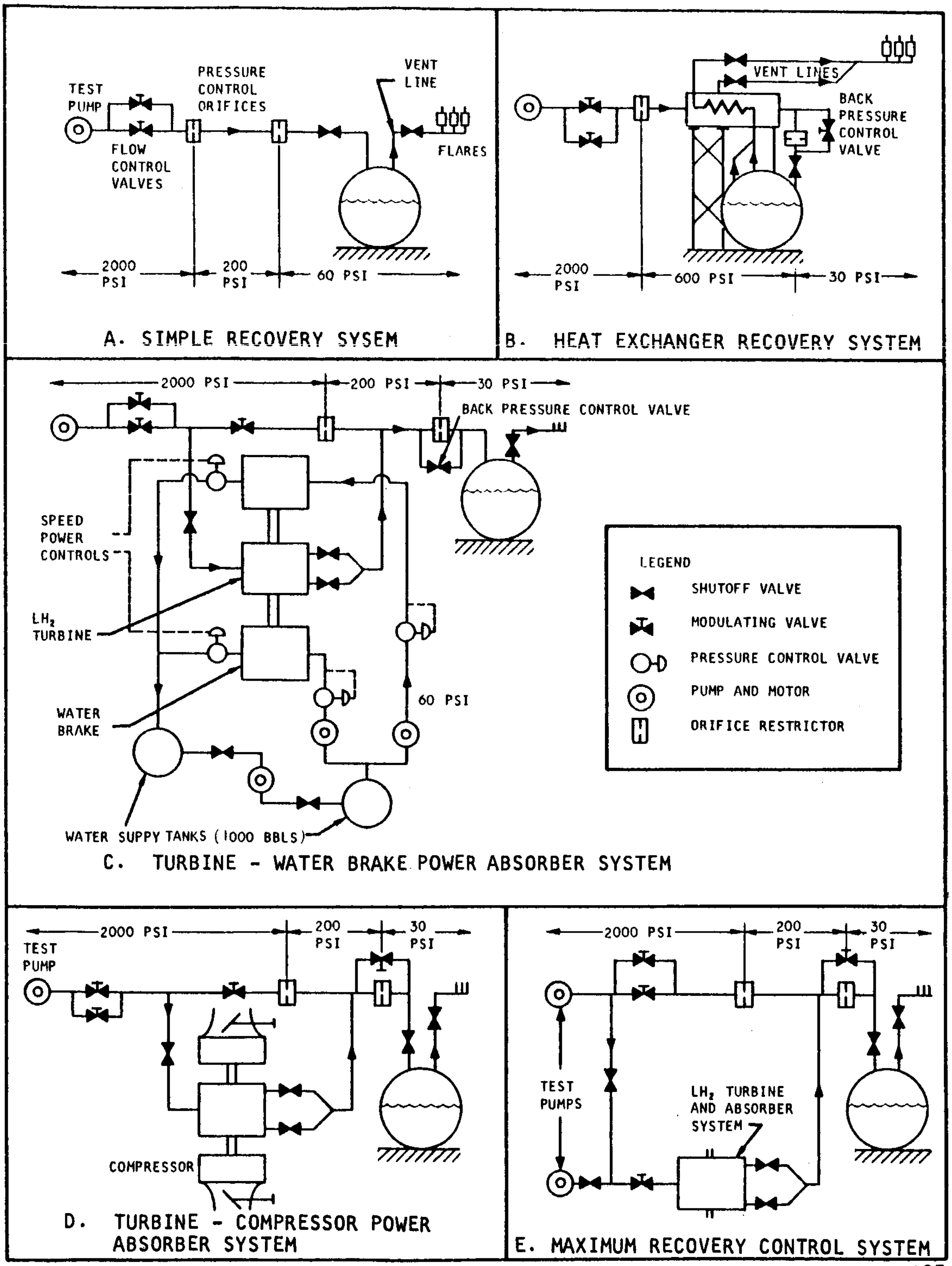

Figure 28. Recovery System Diagrams and Basic Control Elements

Heat Exchanger System

The heat exchanger system has the potential of being the least complex and most reliable, since it is possible to attain a good performance without the use of a bypass or pressure control valves. If the flow rates do not vary appreciably over most of the test period, a simple orifice plate restriction to maintain the appropriate back pressure is an adequate method of control. To obtain a maximum recovery during excursion testing, a control valve in place of the orifice plate is preferable.

For the Aerojet installation, an orifice bypass control valve will be necessary due to the pressure limitations of the 18 inch diameter return line and pressure surges during start up or changes in flow rates. During these transients, a lower than optimum line pressure is necessary. Preliminary estimates by Aerojet engineering indicate that a reserve ΔP = 200 psi may be necessary, which means a maximum back pressure of 400 psi for the heat exchanger during this period. When steady-state conditions have been established, the pressure may then be raised to 600 psi, which is the maximum allowable pressure for the piping, and a "good average" for recovery purposes. Calculations indicate that a small bypass valve, 10 to 12 in. diameter, and a reduced orifice size in the 18 in. return line will be satisfactory.

1. Location

For maximum recovery, the heat exchanger should be located near the tank and in a manner that will minimize the pressure drop requirements of the vent line. The ideal location would be within the tank to eliminate insulation and minimize heat leakage. A more practical location is directly on the tank near the vent outlet. Since the heat exchanger for any system is likely to be massive, this approach means a considerable increase in the load capacity of the outer tank structure and supports, and must be incorporated into the original design.

For the Aerojet system, an offset location on a separate supporting structure would be necessary. The exact geometry will depend upon the final heat exchanger configuration. Figures 29, 30, and 31 show several possible installations for two heat exchanger geometries and locations. These installations are tentative and have not been designed in sufficient detail to indicate an optimum approach.

2. Vent Line Design

Due to the low pressure levels and pressure drops required by this system, the design of the vapor side flow passages in both the heat exchanger and the vent line is critical. From elementary pressure drop relations, a large diameter line is readily apparent. Using the conventional relation,

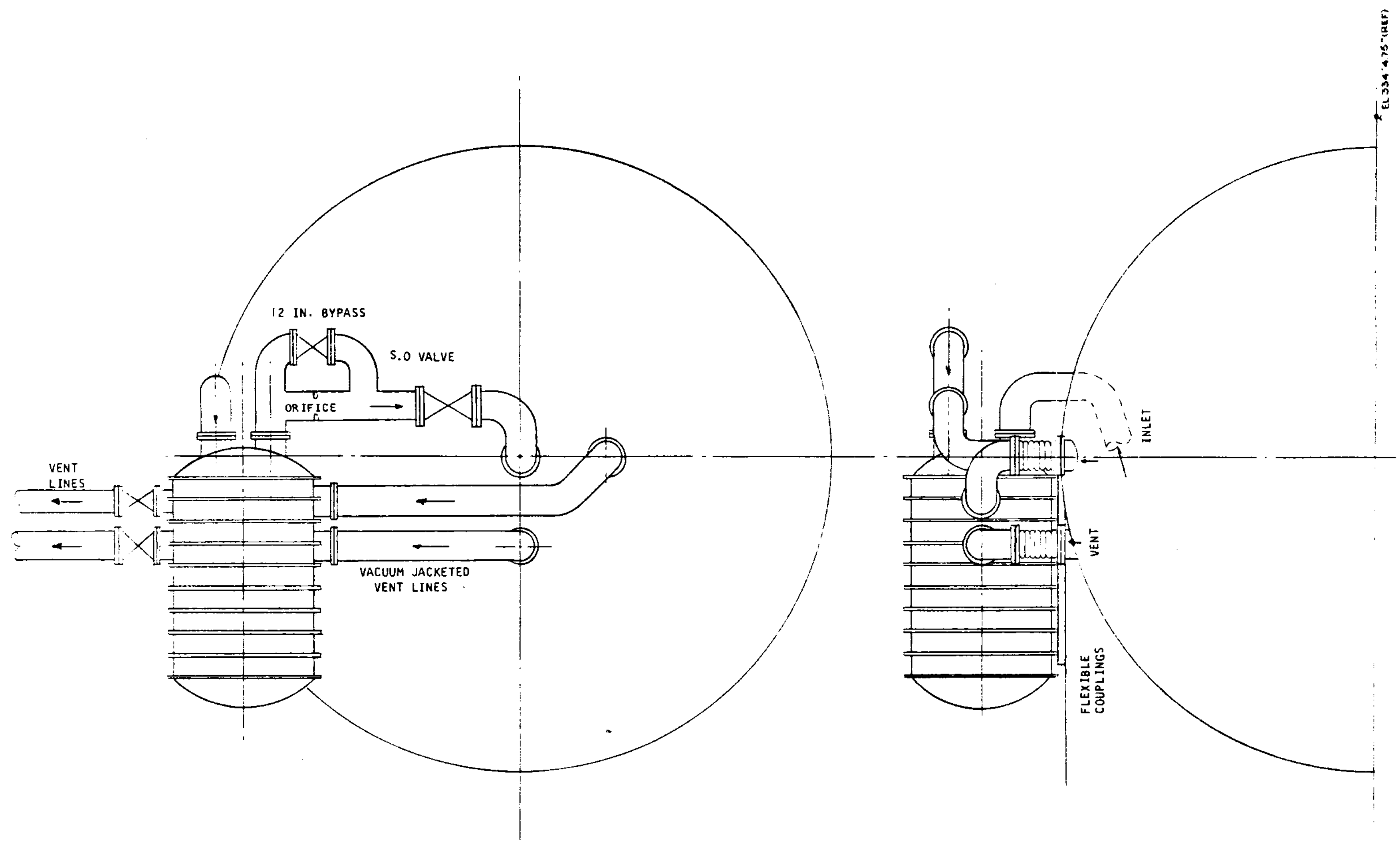

Figure 29. Heat Exchanger Recovery System Installation Aerojet Catch Tank

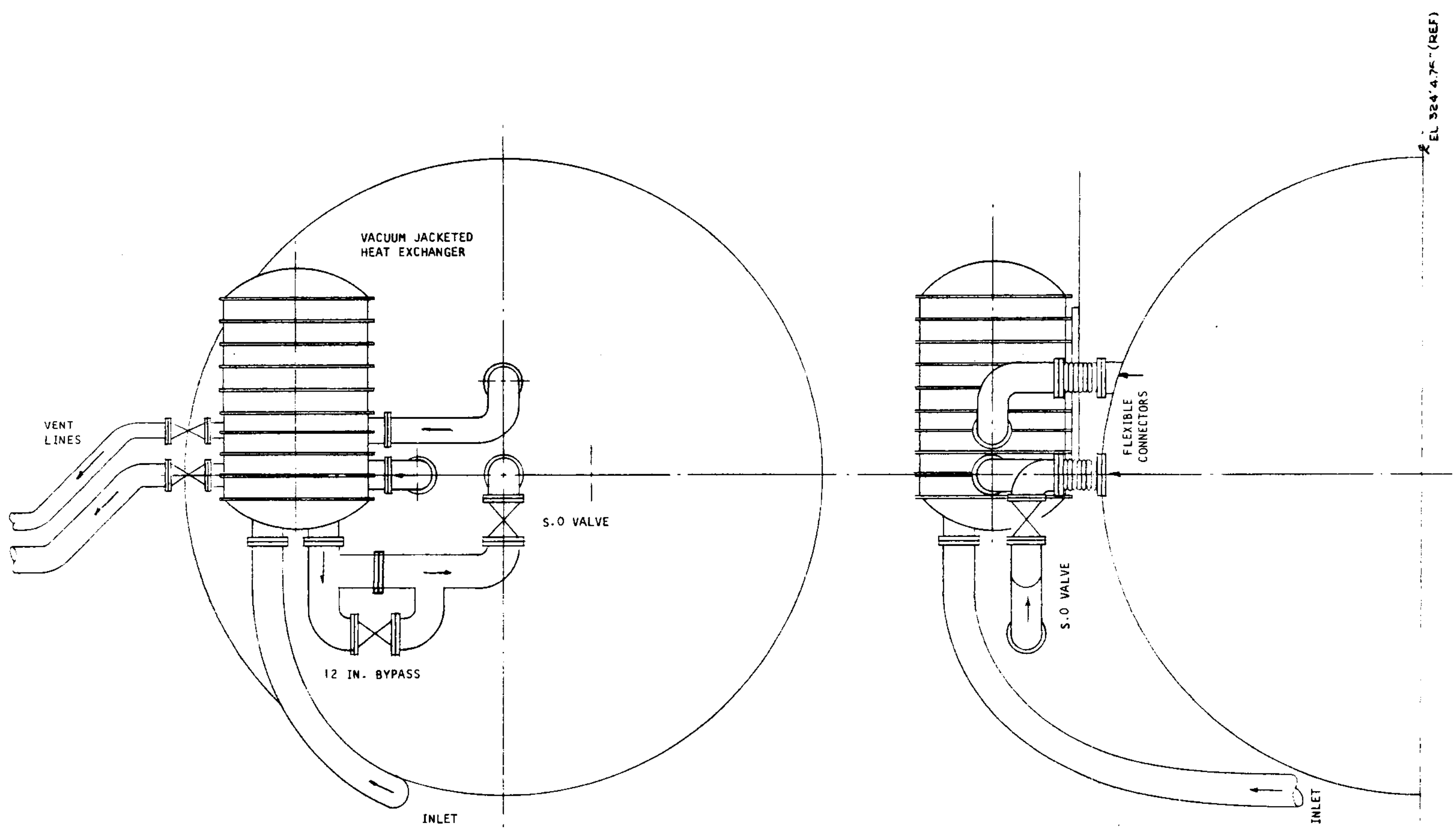

Figure 30. Heat Exchanger Recovery System Alternate Installation

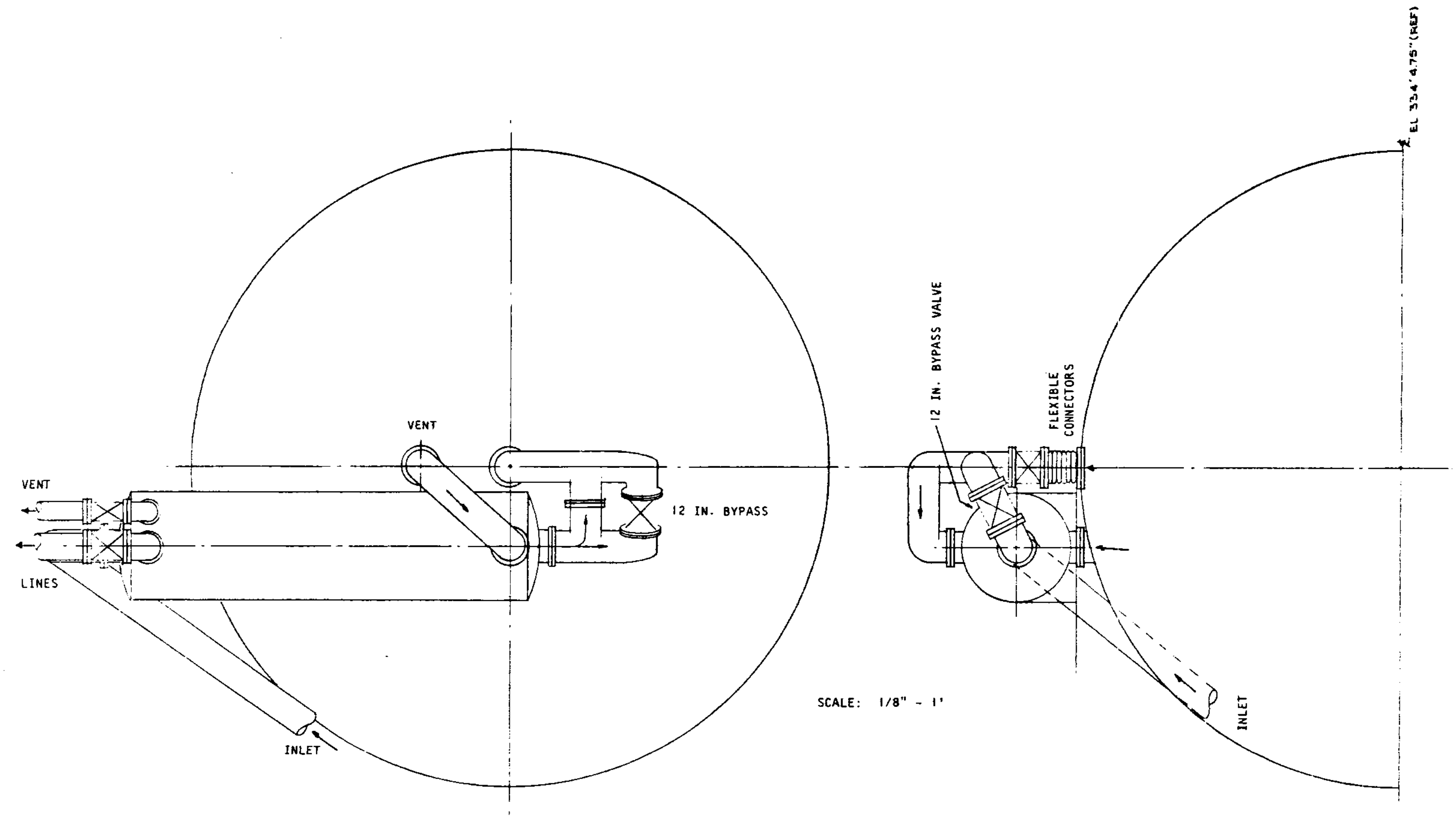

Figure 31. Heat Exchanger Recovery System

$$G = \sqrt{g \frac{P^2 - Po^2}{\Sigma K \; R \; Tav}}$$

$$\frac{A_c}{\dot{W}_v} = .0023 \sqrt{Tav \frac{\Sigma K}{P_r^2 - 1}}$$

Typical values for the above parameters are,

$Tav = 50^{o}R$

$\Sigma K = 3.0$

$P = 20$ psia

$P_o = 14.7$ psia

P_r = pressure ratio, $\frac{P}{P_o}$

$\dot{W}_v = 240$ lb/sec @ m = 60 %

The vent line cross sectional area is then

$A_c = 7.3$ sq ft

and the diameter

$D_v = 3.05$ ft

The value of ΣK for the present Aerojet vent line is 5.5 for an 18 and 20-inch diameter line with 3 flares, as shown in Figure 12. For the maximum flow rate of 330 lb/sec, this loss coefficient indicates a pressure of about 40 psia at the flare exits and a tank pressure about 80 psi. These values appear conservative, since $R_e > 10^8$ and

$$\frac{fL}{D} = \frac{0.004 \times 200}{1.6}$$

$$= 0.50$$

Allowing one velocity head loss for the flares, it appears reasonable that another 1.5 velocity head loss for the bends should be adequate, or a total, $\Sigma K = 3.0$. Thus, a vent line pressure drop of 5.3 psi can be obtained by essentially doubling the diameter of the present line.

In view of the completed vent system at Aerojet, a compromise approach, namely, the addition of a second vent line, was considered more practical. The present supports, which provide for a 16-inch bypass line, can carry a 24-inch line in place of the 16-inch line. Furthermore, a 24-inch shutoff valve is considerably less costly than a single 30 or 36-inch valve. By paralleling a 24-inch line with the present 18-inch line and using the present flow system, a reasonably low pressure drop system is achieved with a minimum of alteration and cost. The line pressure drop for this section will be about 6.5 psi, which will allow an 8 to 10 psi pressure drop for the heat exchanger, when the tank pressure is 30 psia.

3. <u>Heat Exchanger Design</u>

The scope of the present contract permits only a preliminary design of the heat exchanger system, and therefore, a minimum effort on the heat exchanger itselt. From the basic principles established in Section I, certain attributes are immediately apparent, which may be used to obtain preliminary configurations, but not necessarily an optimum design. Obviously, a minimum catch tank pressure dictates a low pressure drop for the vapor side, which in the means relatively low heat transfer coefficients compared to the high pressure side. Therefore, a finned tube geometry is indicated by these relations. Furthermore, a high effectiveness is necessary, which means a counterflow path for the two fluids, and a large surface area. The heat to be transferred for the application is

$$\begin{aligned} q &= \dot{W}_v C_p \Delta T \\ &= \dot{W}_v E_v C_p \,(T\mathrm{max} - T\mathrm{o}) \\ &= 240 \times .90 \times 2.5\,(60 - 42) \\ &= 9700 \text{ Btu/sec} \\ &= 35{,}000{,}000 \text{ Btu/HR} \end{aligned}$$

The corresponding temperature change in the incoming fluid is

$$\Delta T = \frac{9700}{600 \times 3.6} = 4.5^{\circ}, \text{ approximately}$$

The log mean temperature difference is,

$$\Delta T_{lm} = \frac{(60 - 58.2) - (55.5 - 42)}{\ln \frac{1.8}{13.5}}$$

$$= 5.850^{o}R$$

Thus, the overall heat transfer conductance

$$UA = \frac{35,000,000}{5.85}$$

$$= 6,000,000 \text{ Btu/HR } ^{o}F$$

Typical gaseous film coefficients will be about 100-150 and high pressure side coefficients will be 800-1, 000 Btu/HR - FT^2 - ^{o}R, for which U = 80 to 130. Then the heat exchanger surface area, A_S, = 75,000 to 46,000 sq ft.

It is generally desirable to minimize the number of high pressure tubes by increasing the vapor side area through the use of fins. This method results in a more compact heat exchanger, which is desirable from the heat leakage standpoint between runs. Using currently available data, a typical design results in the following values:

Requirements

	Hot Side	Cold Side
Fluid	Liquid H_2	Vapor H_2
Flow Rate, lb/sec	600	250
Inlet Temperature, ^{o}R	60	42
Inlet Pressure, psia	*	20
Pressure Drop, psi	*	10
Effectiveness	*	0.90,

Typical Heat Exchanger Core	Hot Side (Inside Tube)	Cold Side
Fluid	Liquid H_2	Vapor H_2
Effectiveness, E	0.32	0.90
Mass Velocity. G, lf/sec-ft^2	179	6.68
Heat transfer coefficient, h, Btu/hr-ft^2 -°R	1752	143
Heat transfer surface area, A_t, ft^2	16,900	163,000
Fin effectiveness, η_f	———	.769
Total core pressure loss, ΔP core, psi	6.86	7.44
*Viscosity of the fluids, μ, lb/ft-sec x 10^6	5.78	1
*Thermoconductivity of the fluids, k, B/hr-ft-°F	0.033	0.0125
*Specific heat of the fluids, C_p, B/lb-°F	3.9	2.91
*Prandtl number, Pr, of the fluids	2.00	0.76
*Density of the fluids, ρ, lb/ft^2	3.8	0.1
Tube length, ft	35.7	
Number of tubes	5320	
Number of fins per inch	8.72	
Tube outside diameter, inch	0.38	
Tube wall thickness, inch	0.020	
Fin outside diameter, inch	0.92	
Fin thickness, inch	0.018	

*Based on core average temperature and average pressure.

4. Heat Exchanger Geometry

With respect to the heat exchanger geometry, a certain latitude exists because of the trade-off relations between surface area and heat transfer coefficients. Since the vapor side coefficient is controlling, the Reynolds analogy between heat transfer and the friction factor results in the following approximate relation:

$$\frac{A_{sv}}{N} A_{cv} = \frac{4\dot{W}^2 \Delta T_v}{g \rho_v \Delta P_v \Delta T_{lm}}$$

where A_{sv} = surface area, vapor side

A_{cv} = cross-sectional flow area, vapor side

N = number of tube rows

$\dot{W}$ = flow rate

ΔT_v = vapor side temperature rise

ΔT_{lm} = log mean temperature difference

ρ_v = vapor density

ΔP_v = pressure drop, vapor side

The variables on the right-hand side are determined by the desired recovery and become a constant for a certain design point. Then, the free flow area, A_{cv}, the surface area, A_{sv}, and the number of tube rows, N, can be varied to give equivalent results in terms of pressure and temperature drops through the heat exchanger. This relation permits a considerable variation in the design geometry and indicates a possible use of available equipment. Also, if the surface and cross-sectional areas are overdesigned, the recovery performance is improved, as indicated by a lower ΔP and a higher ΔT_v.

5. Vacuum Jacketing

As discussed in the cooldown analysis, vacuum jacketing is considered necessary. Figure 32 shows the geometry which will minimize the surface area and the size of the vacuum jacket. The heat exchanger cores are arranged into two banks separated by a transverse baffle at the vapor inlet and outlet ports. This configuration shortens the length of the individual cores to obtain a better length/diameter radio than a straight tube and shell arrangement.

The jacket, as shown in Figure 32, will be a cylindrical shell about 12 feet in diameter and 25 feet long, with sufficient strength to support the core as shown under high vacuum pressure. The shell is supported by I-beam rings to reduce weight and material costs. Although elliptical heads are shown for the ends, beam-supported flat plates may be a better approach for this size structure.

6. Performance

The performance of this system has been discussed previously under recovery thermodynamics, and the comparison with the throttling system was given in Figure 5. The dashed line shows the estimated performance for the M-1 test pump range of operating points as given in Figure C-1. The slightly reduced performance is due to the pressure limitation of the existing lines which limit the maximum pressure to 600 psi.

Turbine Recovery System

The recommended turbine recovery system will consist of the axial turbine, the air compressor loader and the control system which is diagrammed in Figure 28 D, and which is shown in the layout drawing Figure 33. The turbine is located at the present position of the upstream orifice in the Aerojet return line and requires the construction of a bypass for starting and off design point operation. Using currently available hardware (Hammel-Dahl valves), the installation would have the configuration shown in the drawing. Other configurations, using ball shutoff valves at the turbine inlet may be preferable to reduce pressure drops and may also reduce installation costs as a result of fewer bends in the vacuum jacketed lines. These aspects of design were not optimized in this study. Not shown in the drawing is the orifice bypass valve at the catch tank. This valve will be of the same size as that shown for the heat exchanger system and will be used in conjunction with the downstream orifice to maintain the appropriate turbine back pressure for various flow conditions.

Three shutoff or isolation valves are shown, one 12-inch high pressure valve ahead of the turbine and two 12-inch low pressure valves in the discharge. The reason for the two 12-inch valves instead of one 18-inch valve is cost, based upon preliminary estimates.

Right angle valves are shown for the upstream position because of the need for a balanced valve construction where a high pressure drop exists. This requirement is particularly true for the modulating valve in the by-pass line, but may not be required for the shutoff

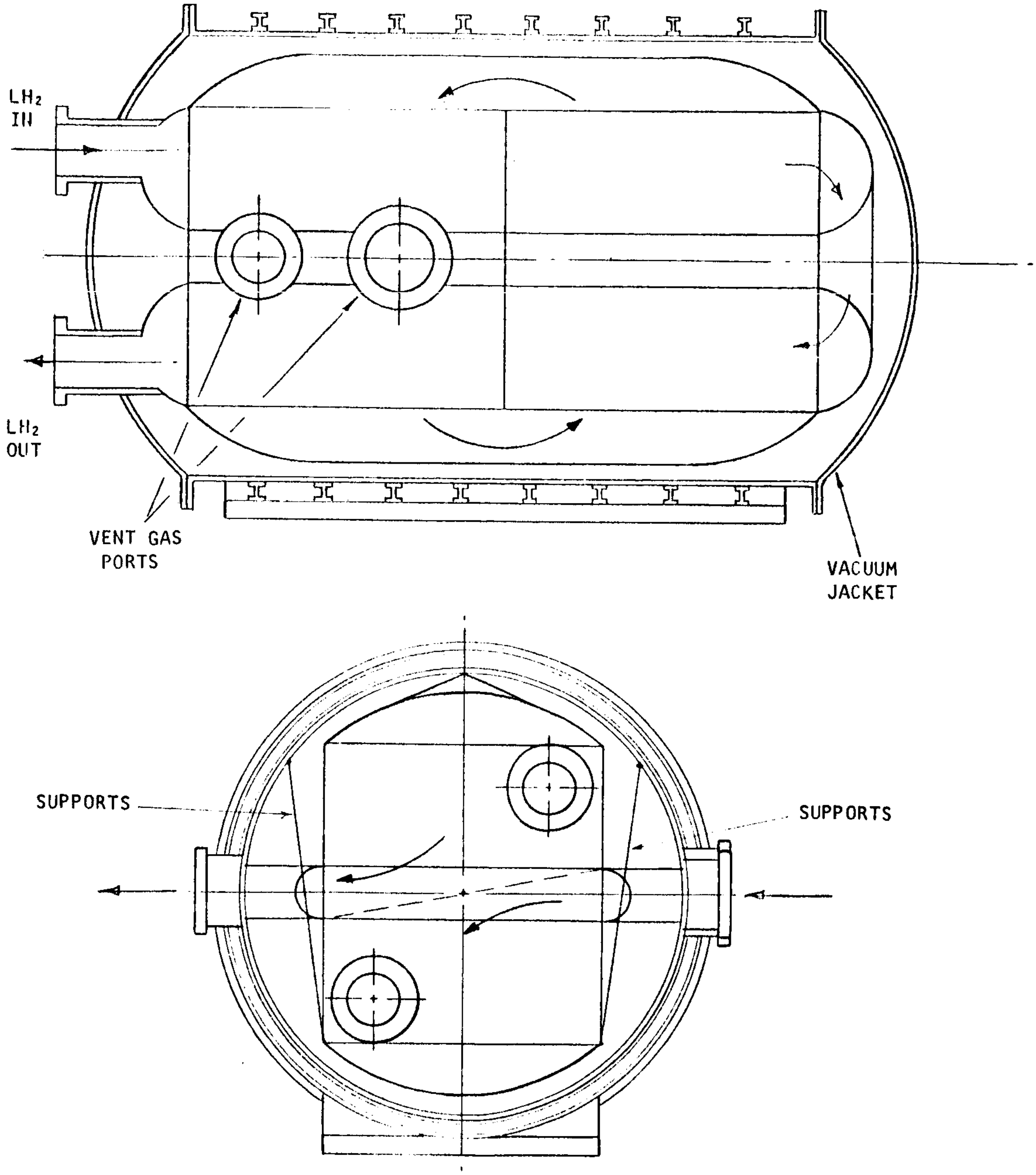

Figure 32. Heat Exchanger

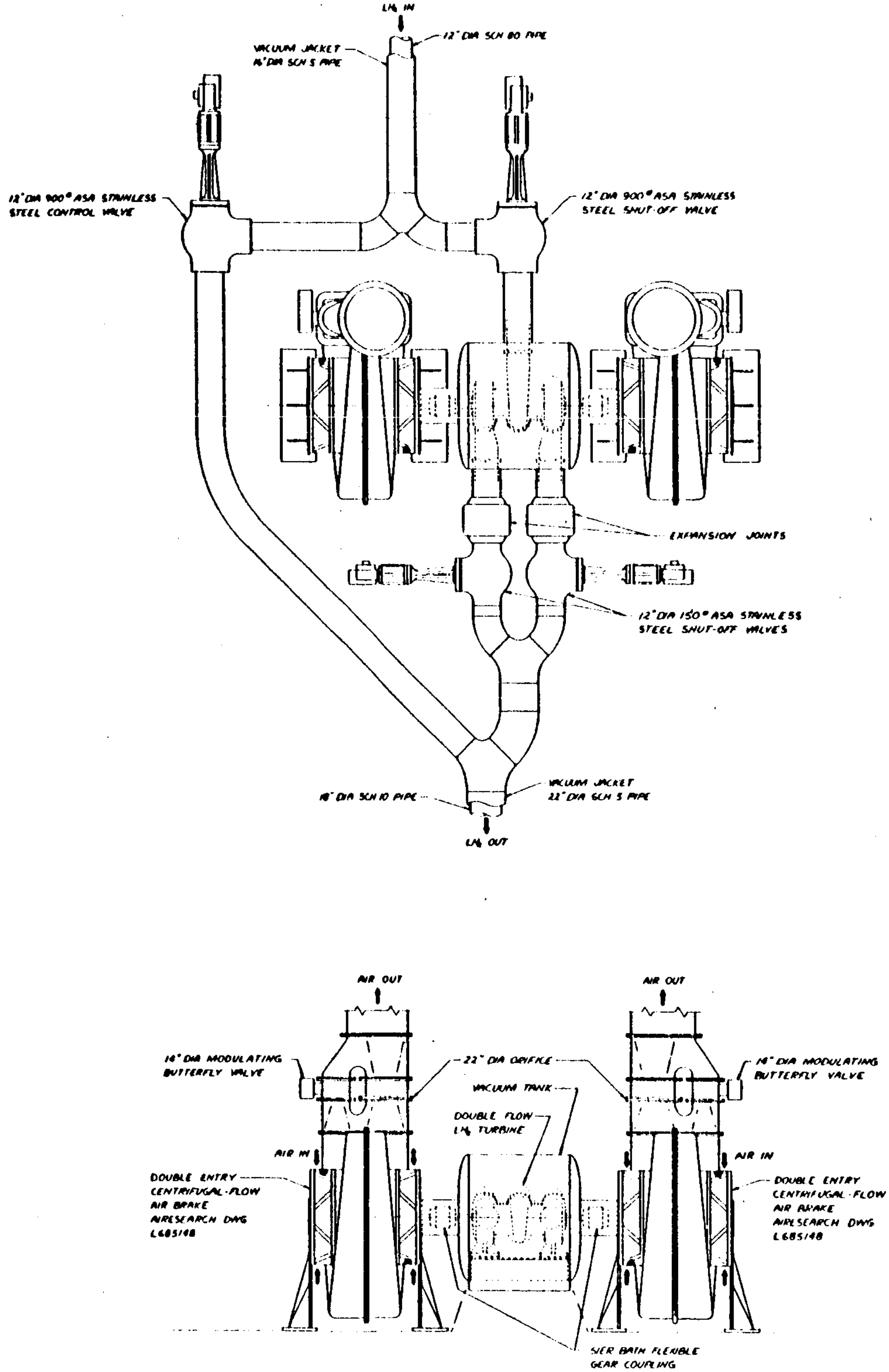

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Figure 33. Optimum LH_2 Recovery System Cryogenic Turbine and Centrifugal Air Brakes

valve. At this time, the pressures drop across this valve will be relatively low.

The installation will consist of vacuum jacketed lines with expansion joints in the low pressure discharge lines. The turbine will also be vacuum jacketed.

The valves will be electro-hydraulically actuated, using the 3000 psi hydraulic supply in the area. Individual accumulators and filters will be provided. Details of the response characteristics of the valves and other control system elements are given in the next section.

1. Operation and Performance

The turbine operating line for a constant back pressure of 180 psia is shown superimposed on the estimated pump characteristics in Figure 34.

For the aforementioned back pressure, the turbine speed varies from a minimum of 0.91 of design speed at a pump speed of 11,900 rpm to a maximum of 1.135 at design speed at a pump speed of 14,550 rpm. Over this range of rotating speed, the turbine efficiency will remain largely constant, hence the 180 psia back pressure line is a line of optimum hydrogen recovery for the axial turboexpander system.

For a given pump operating speed, as the delivery head is increased to values which are higher than those indicated by the "optimum" line, the delivery flow decreases. Since the pump is matched to a fixed geometry turbine, the turbine inlet pressure must be throttled to keep the turbine head compatible with the decreasing pump flow. At the minimum pump operating speed of 11,900 rpm, the turbine inlet pressure will be reduced to a minimum of 1140 psia, for a turbine speed of 5450 rpm.

The cumulative effect of inlet pressure throttling is to reduce the hydrogen recovery rates somewhat below those of a turbine whose nozzle can be varied to efficiently match the flow at the higher head. It is to be noted, however, that sizeable improvements over and above simple throttling are still everywhere in evidence.

For a given pump speed, as the pump head is reduced relative to those indicated by the "optimum" recovery line, the quantity of liquid hydrogen delivered by the pump increases. In order to handle the higher flow rate, the turbine head must be increased. Since the pump delivery head is decreasing, the back pressure to the turbine must be rapidly decreased to provide the flow match. The turbine will temporarily speed up. Unfortunately, the decrease in turbine back pressure is limited by the onset of cavitation in the last stage rotor. As indicated by the cavitation study, the limiting back pressure is 100 psia.

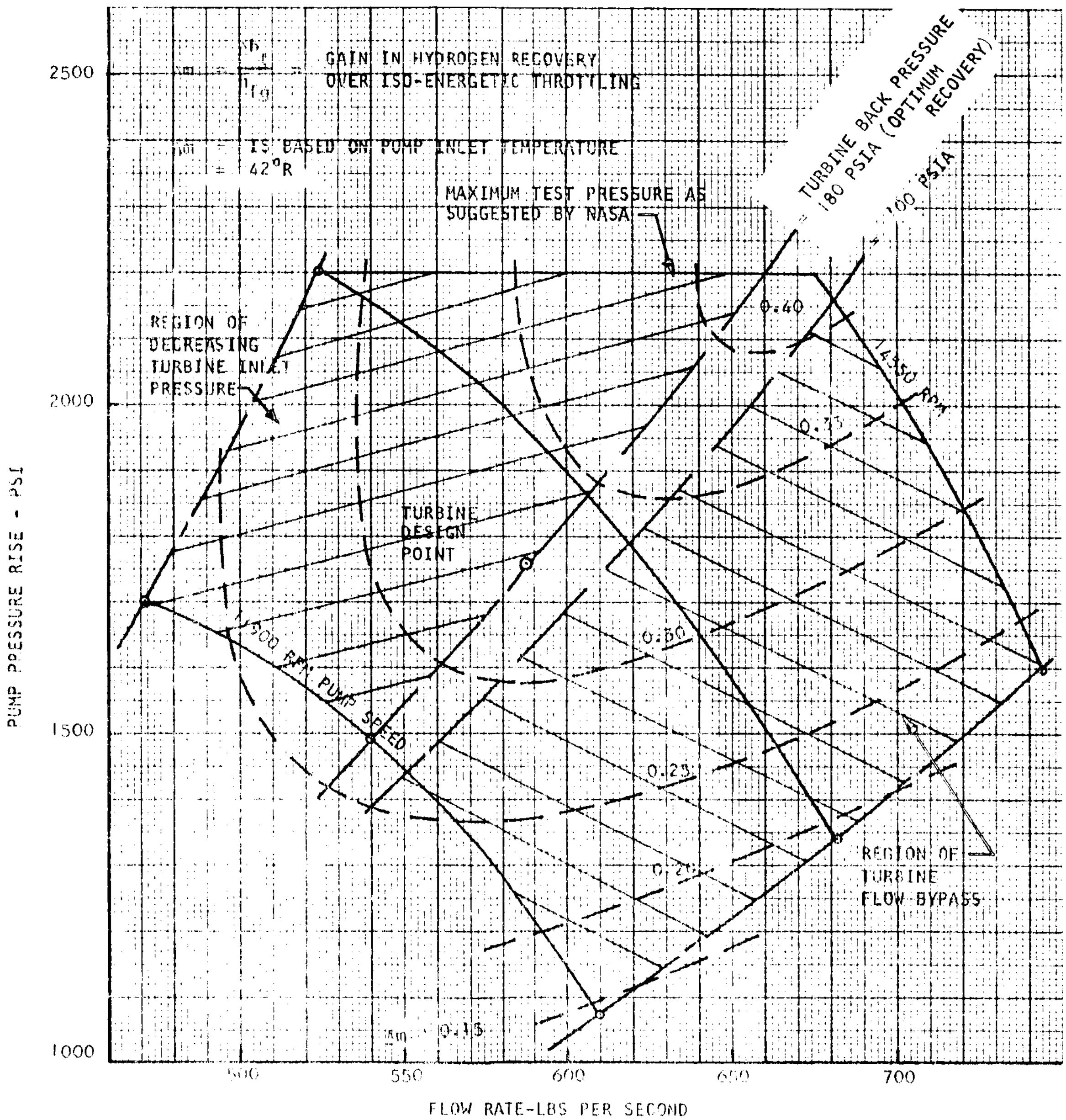


Figure 34. Estimated Performance of Double Flow Axial Turbine and Aerojet LH_2 Pump

VI CONTROL SYSTEM STUDY

From Section I - Recovery System Thermodynamics, the basic operation criteria for a successful turbine recovery system may be established. Namely, for any given energy level of the fluid at the pump outlet, the amount of liquid returned to the catch tank may be maximized by extracting maximum turbine work. Since pressure loss in the flow line influences the available energy for conversion to work, maximum turbine work may be extracted only if the turbine is operated at peak efficiency and line losses are reduced to a minimum.

Control Requirements

The controls for the turbine recovery system must be sufficient to satisfy the following modes of operation.

(1) Transition period when the turboexpander begins operation

(2) Steady-state operation for a given pump test

(3) Failure mode operation (system protection)

Control requirements of each of the above operational modes will be discussed in detail in the sections to follow, but first, the basic pump test configuration will be considered. The basic pump test system is as shown in Figure 35.

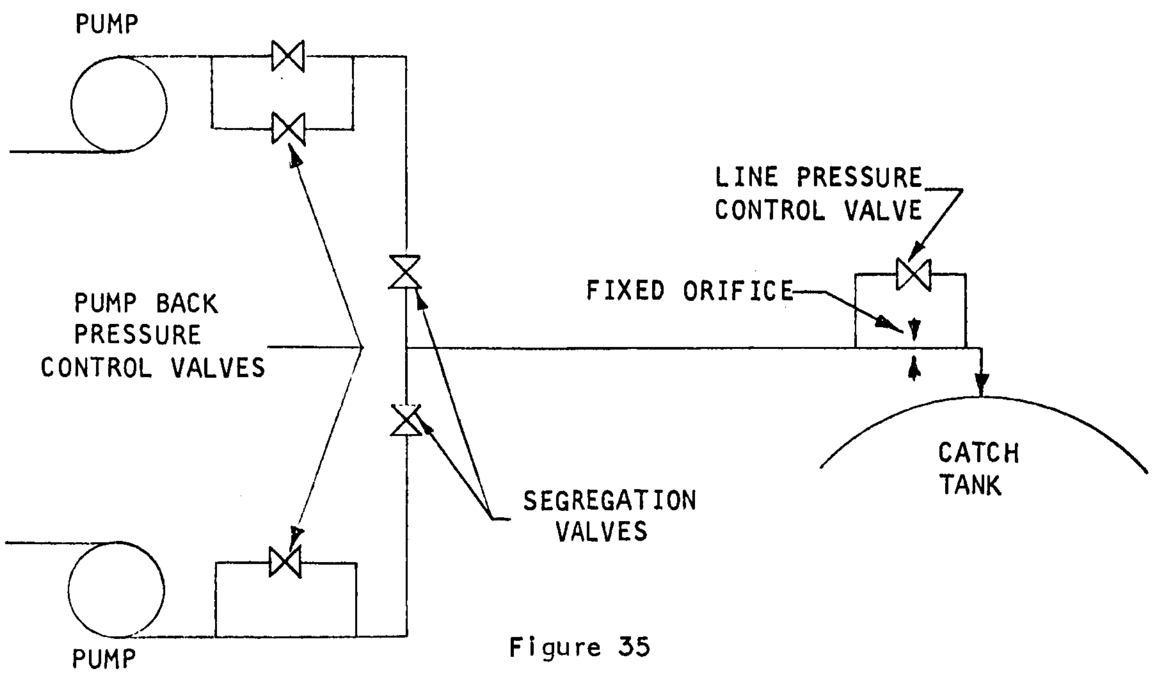


Figure 35

If the pump head is further decreased (resulting in a continuing increase in delivery flow), the pump flow will exceed the capacity of the turbine which now must operate at a constant back pressure. The flow surplus is then bypassed to the simple throttling line.

The bypass flow will not be constant but will increase due to the fact that the turbine head will start falling off as the pump continues to be throttled.

The turbine rotating speed will begin to reduce after the back pressure is established at 100 psia, and for the maximum pump speed of 14,550 rpm it reaches a minimum of 6460 rpm. At the minimum pump speed of 11,900 rpm, the turbine will decelerate to 5210 rpm.

As in the case of the turbine running at throttled inlet pressure, the potential hydrogen recovery rates possible for an efficient variable geometry unit are not realized. However, the estimated performance is still markedly superior to simple throttling.

Estimated incremental hydrogen recovery rates have been calculated for the entire pump operating spectrum, using the previously defined formula:

$$\Delta m = \frac{\Delta h_t}{h_{fg}}$$

and making the proper allowances for flow bypass in the low pressure head region of pump operation. The results of this calculation have been plotted as recovery contours on Figure 34. It is to be noted that all recovery calculations have been made assuming a pump inlet temperature of $42^{\circ}R$.

The turbine reaches a maximum speed and output power of 7820 rpm and 79,800 hp, respectively, at the zero flow bypass and minimum back pressure point for the pump speed of 14,550 rpm.

The minimum turbine speed and output power for steady-state operation occurs at the maximum pump flow point at a pump speed of 11,900 rpm. The values are 5210 rpm and 23,750 hp, respectively.

During the transition period, all changes in valve position must be made slowly to prevent high pressure fluctuations (water hammer) from affecting the test pump operation.

Steady-State Mode

Steady-state speed of the turbine recovery system will be regulated by reducing accelerating torques to zero. This is accomplished by changing the absorbed power to match the power extracted by the turbine. For either the air compressor or the water brake loader, the power absorbed may be varied by a control valve which changes the restriction in the fluid outlet line. The speed control itself may consist of standard electronic and hydraulic components. The generalized speed control configuration is as shown in Figure 37.

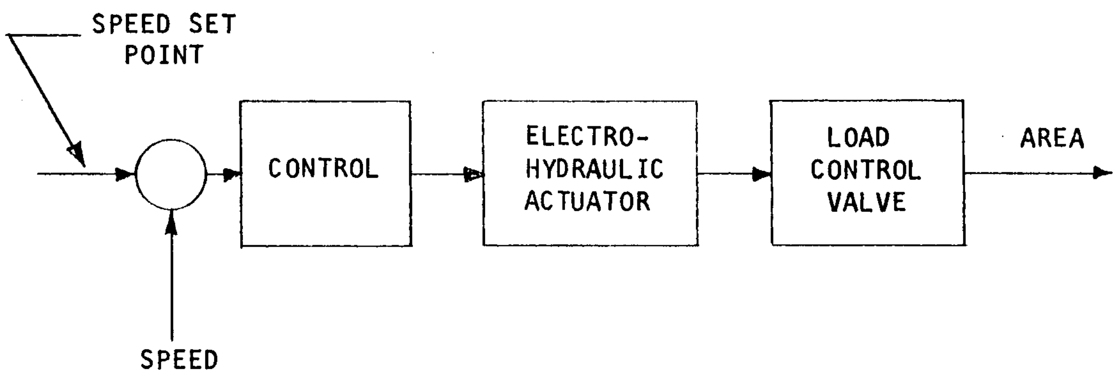


Figure 37

For best overall recovery system performance, thermodynamic considerations dictate that the turbine back pressure be the minimum that will avoid cavitation. This requires the paralleling of the fixed orifice at the catch tank with a pressure control valve. However, for a given pump test, turbine work must be maximized independent of its back pressure. It may be shown (Appendix D) that the turbine work may be approximated as

$$HP = K_1 \Delta P_t N - K_2 (\Delta P_t)^{1/2} N_2$$

where HP = turbine power

P_t = turbine pressure differential

N = turbine speed

The pump back pressure valves are used to establish a given pump test (i. e. , steady-state and transient simulation). Segregation valves permit operation of either one of the two pumps with a common catch tank. To suppress boiling in the line to the catch tank, line pressure is maintained sufficiently high by an orifice and control valve near the catch tank inlet.

Transition Mode

During pump startup and transient simulation, it will be necessary to isolate the recovery turbine from the system so that there is no influence upon the pump test from the recovery system. A shutoff valve on either side of the turbine will isolate it during this period of operation. To initiate operation of the turboexpander when steady-state pump testing commences, it is necessary to divert the flow to the catch tank through the turbine isolation valves and flow diversion valve which satisfy the above principle are as shown in Figure 36.

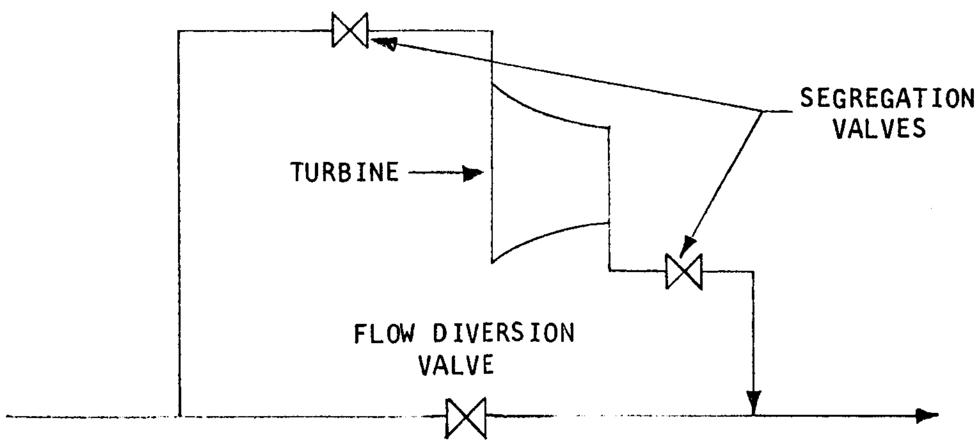


Figure 36

K_1, K_2 = proportionality constants

Differentiating (2) with respect to N and equating to zero yields the following relationship which defines the necessary conditions for peak power

$$N = K_3\sqrt{\Delta P_t}; \; K_3 = K_1/2K_2. \qquad (3)$$

Maintaining the turbine speed in proportion to the square root of the turbine pressure differential will ensure peak power extraction over the entire pump operating map.

In summary, controls necessary for the steady-state operation of the turbine recovery system are:

(1) A line pressure control at the catch tank inlet

(2) Speed sensor-control-load valve assembly

(3) Scheduling device which changes the speed set point in proportion to the square root of the turbine differential pressure

(4) Method of adjusting the restriction as seen by the pump under test

Failure Mode Controls

Two possible failures that may be experienced with the turbine recovery system are (1) seal failure where hydrogen could escape to the atmosphere, and (2) loss of load or associated speed control apparatus. Failures of the leakage type are isolated from the system by simply closing the segregation valves on either side of the turbine. It is possible to operate the test facility without the turbine recovery system until such time that appropriate repairs may be completed. Loss of load will cause the turbine to overspeed. If the absorber drive shaft were to break, the absorber drive shaft were to break, the absorber will slow to zero speed at a rate given by the absorbed load. The turbine will accelerate to runaway (no load) speed which is well within design limits. Overspeeds resulting from loss of control apparatus are eliminated by the speed control design. The load control valve will be designed such that loss of signal will result in automatic opening or closure (depending upon the load absorption technique), thus increasing the absorbed load and reducing speed. No additional control elements are necessary for failure mode protection of this system.

Control System Description

Control requirements as described in the previous section dictate the system configuration. There are essentially two approaches which may be considered since transient and steady-state simulation valves are placed in the flow line directly downstream of the pump. These valves are in parallel and have a total $C_v(H_2) = 6025$. At the design point (600 lb/sec flow rate), the pressure drop is given by

$$\Delta P = \left[\frac{Q}{C_v(H_2)}\right]^2 = \left[\frac{61,200 \text{ gpm}}{6025}\right]^2 = 103 \text{ psi}$$

which represents a loss of \$1,450,000 in potential recovery. Since the pressure drop across these valves is quite high, it would be desirable to connect the turbine recovery system upstream of the transient and back pressure valves, and thus operate the recovery system at minimum line loss. Connecting the recovery system between the pump outlet and associated back pressure control valves will undoubtedly require modifications in the present system and possibly delay completion of the facility. Therefore, a description of control system configuration and operation of each of the two alternatives will be presented.

1. Ultimate Recovery System

The system configuration illustrating locations of all control components with an air compressor as a load absorber is presented in Figure 38. (A system utilizing a water brake power absorbtion system is identical in all respects to the air compressor system except valves are placed on the water inlet and outlet lines, with the outlet valve modulated as a function of speed error.) Noting the respective locations of all remote segregation valves, it is obvious that they simply allow changes in the hydrogen flow path such that two pumps may be used in conjunction with a common recovery and catch tank system. For purposes of discussion, it shall be sufficient to consider the operation of only one pump.

a. Transition Mode--During pump startup and transient simulation, segregation valves are closed to isolate the recovery system.

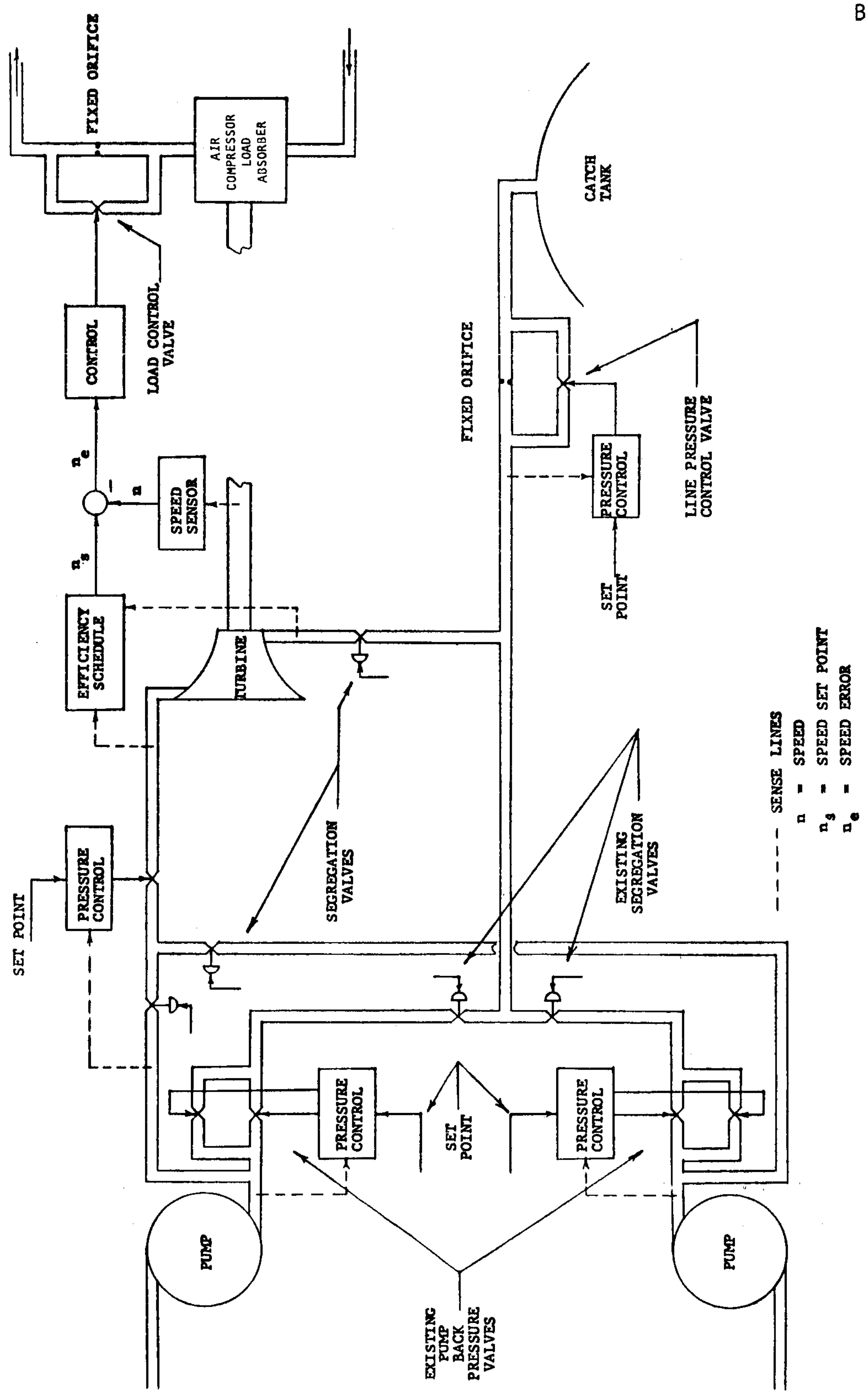


Figure 38. Ultimate Hydrogen Recovery System

The line pressure control valve maintains a prescribed line pressure or can be manually fixed if desired and the pump back pressure control valves operate in accordance with a predetermined program. Initiation of recovery system operation commences upon attainment of steady-state operation of the pump. If a scheduled area start is made, the back pressure valve may be switched to automatic pressure control after the completion of the transient start. At this time, the remote set point of the pressure control valve immediately upstream of the turbine (turbine inlet valve) is set sufficiently high so the valve remains closed. Seregation valves on either side of the turbine are opened, then the turbine inlet valve is opened by lowering its set point at a constant rate which avoids water hammer. Back pressure control valves at the pump outlet close automatically in an attempt to preserve the pressure set point. To avoid disturbances in pump back pressure, the turbine inlet and pump back pressure valves will be sequenced as shown in Figure 39 during the transition period.

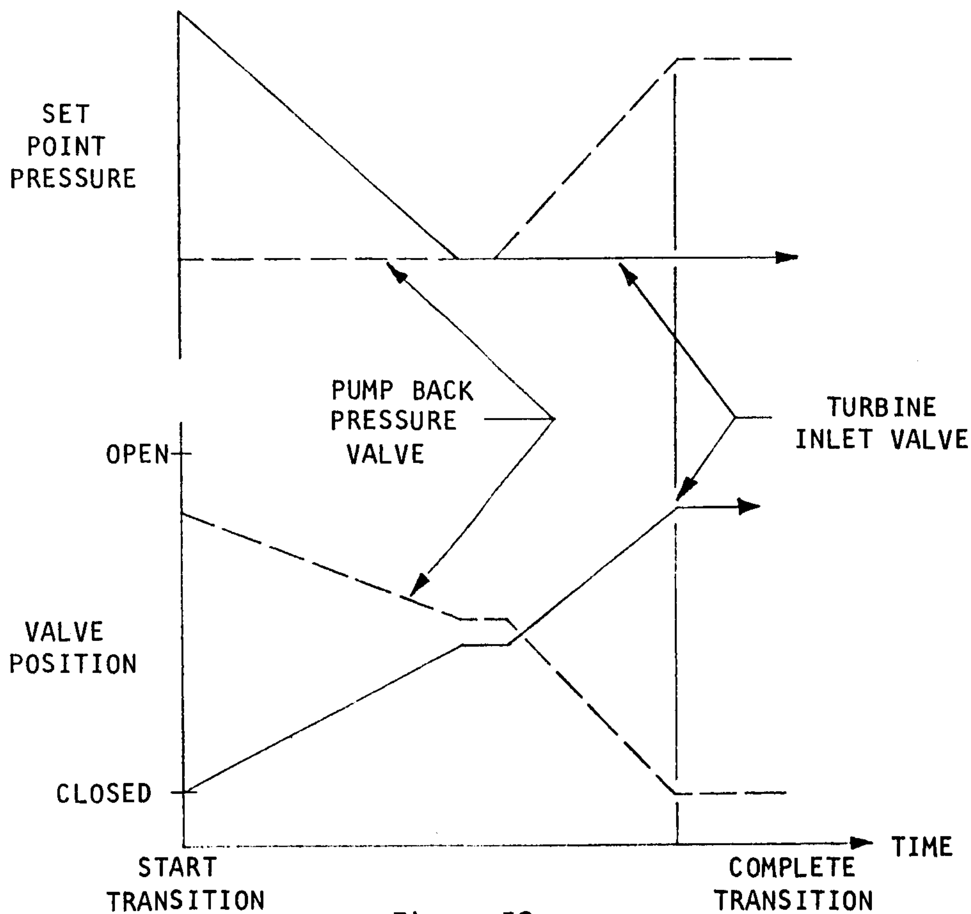


Figure 39

When the transition period is complete, the turbine inlet valve functions as the pump back pressure control. The original control valves remain closed unless specific tests (low pump back pressure) require bypass flow.

Figure 40 following, shows a block diagram of the speed control during transition. In brief, the speed control is de-energized which opens the load control valve and a fixed area representing more than design load is placed in the absorber fluid outlet line. During transition the water brake system would respond in a similar manner consisttent with the above principle that greater than design load is applied when the speed control is de-energized. When the recovery system reaches steady state (pump back pressure valve closed) the speed control is energized and the load valve area moves in accordance to instructions received from the speed control to bring turbine speed to the steady-state design point.

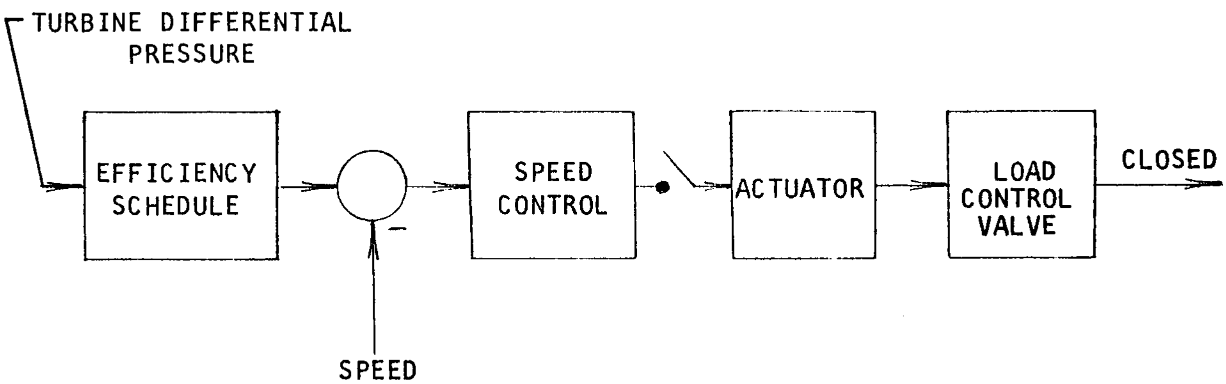


Figure 40

b. Steady-State Mode--In steady state the turbine inlet valve modulates to maintain the pump back pressure at a prescribed setting, and the speed control changes the load valve to maintain the correct turbine speed. The turbine inlet valve has a self-contained regulation system and need not be discussed further in this section. However, the turbine speed control shall be considered in further detail. Refer to Figure 40 with the speed control switch energized.

In principle, turbine speed is synthesized from the rotating group, compared with the set or desired speed resulting in a speed error signal. The control senses speed error and sends a signal to the valve actuator which reduces the error by opening or closing the load control valve. Turbine power or work is a function of the square root of the turbine pressure differential ($\sqrt{\Delta P_t}$). The efficiency schedule is a device which manipulates the speed set point proportional to $\sqrt{\Delta P_t}$. This device is necessary to maximize the recovery over the entire operational range of the test pump. Actual mechanization of speed control hardware will be dicussed in a later section.

In the event pump testing must occur at sufficiently low values which require more than full flow through the turbine, the original pump back pressure valve is put on pure manual control and opened to a predetermined setting. The turbine inlet valve will then modulate the flow to maintain the "low" set point pressure. The original turbine back pressure valve is functioning as a bypass valve for this type of test.

c. Failure Mode--It has been shown that requirements for failure mode protection do not require additional control elements. In the event of hydrogen leakage at the turbine, turbine segregation valves will be closed and the pump back pressure valve set point will simultaneously be lowered to the required value.

During this operation, the pump back pressure will not change from the control point and the pump test may continue without the recovery system in operation.

Failures of any member of the speed control apparatus will result in not more than 100 percent overspeed which the rotating group will be designed to accommodate. The pump test may continue as normal without the benefit of recovery.

2. Standard Recovery System

The system under consideration is as illustrated in Figure 41. This configuration tends to separate the test pump and recovery systems. By proper system design, the total restriction downstream of the pump back pressure valves will remain the same when the recovery system is in or out of operation, thus eliminating system interaction between the recovery system and the pump. As previously discussed, segregation valves allow the use of two pumps in combination with one recovery system and catch tank. Operation of the control system is much the same as the minimum pressure drop system discussed above and only differences will be pointed out.

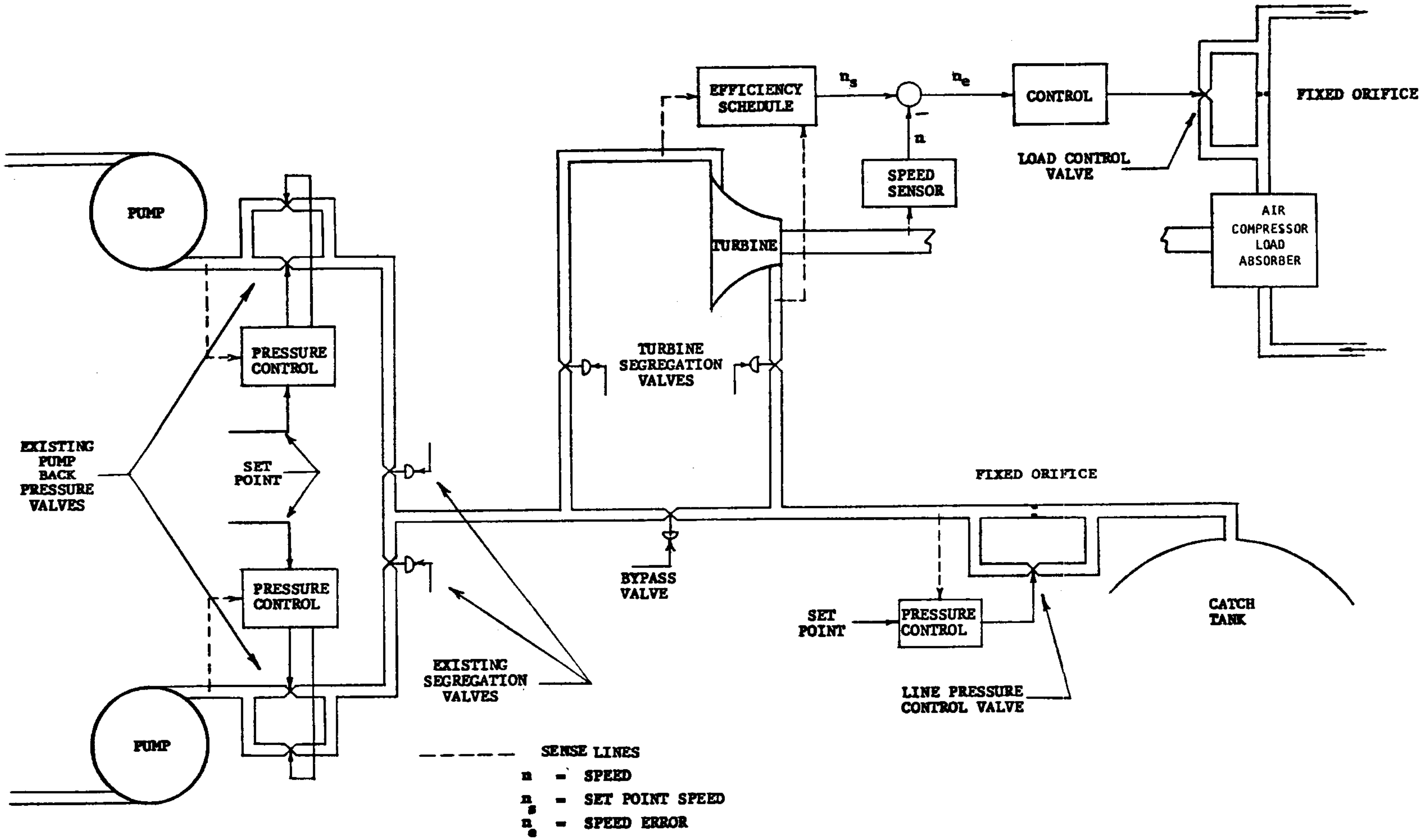


Figure 41. Standard Hydrogen Recovery System

a. Transition Mode--Turbine segregation valves are closed until the pump transients are over. At all times the line pressure control maintains a predetermined line pressure. Operation of the turbine recovery system is initiated as the turbine segregation valves are opened followed by closing of the bypass valve. The pump back pressure valves function as normal to maintain the proper pump outlet pressure.

b. Steady-State Mode--Operation of controls during steady state are identical, with one exception described in the minimum pressure drop system. In this configuration, the turbine bypass valve is a pure bypass valve and is merely opened manually to enable the pump back pressure valves to maintain control at a reduced set point pressure.

c. Failure Mode--Overspeeds are accounted for by the system design as described in the previous configuration and leakage type failures are compensated for by closing of the turbine segregation valves and simultaneously opening the bypass valve. Opening and closing rates of these valves shall be programmed so that little or no back pressure changes are experienced by the test pump.

Component Specification

In the previous portions of this report two recovery system configurations and the associated control logic have been discussed. A dynamic analysis of the system has been presented in Appendix D, thus completing the requisites necessary for component specification.

In general, the control components specified are to be interchangeable with existing components in the Aerojet system. The valves are actuated with 3000 psi hydraulic actuators and electric feedback to the control room. Electronic analog computor type amplifiers compute the signal sent to the hydraulic pilot valves.

Elements Common to Both System Configurations

This section will be concerned with the speed-load absorption control, line pressure control, and turbine back pressure segregation valve.

(1) Speed Control--The speed control shall consist of (1) efficiency schedule, (2) speed sensor, and (3) a load control valve which incorporates integral control action. Load control valve sizing will be included

for an air compressor only as a water brake would be purchased complete with control apparatus. However, the control mode and associated specifications will be specified for a water brake installation.

The efficiency schedule apparatus will consist of pressure transducers to measure the turbine pressure differential (2680 psi maximum upstream and 100 to 600 psi downstream) and a solid state electronic network designed such that the electrical output signal (speed set point) is proportional to $\sqrt{\Delta P_t}$. Turbine speed will be sensed and also converted to an electrical signal. The load control valves (air compressor or water brake) shall be constructed so they are spring and pressure loaded so as to increase the load in the event of signal loss. The maximum rate of integral control action for load control valves are given in Appendix D. The compressor shall be loaded by an orifice plus full valve area to achieve the maximum loading. To decrease the loading, flow area is decreased by closing the valve. Preliminary physical sizing is given below:

Orifice: 22 in. diameter - sharp edged (may be revised when compressor maps are complete)

Valve: 14 in. butterfly valve (assuming 25% blockage)

Response of load control valves for each type absorber should be in the order of 3 or 4 seconds for full valve travel.

(2) Line Back Pressure Control--The control configuration is a fixed orifice paralleled by a control valve as shown in Figures 38 and 41.

The valve and orifice have been sized to control line pressure from 100 psia to 600 psia over the pump operational range as given in Figure 13 Sizing is as follows:

Orifice: 5.4-in. diameter sharp edge

Control Valve: 12-in. remote set, hydraullically actuated, integral control valve. This valve is of the same type used for pump back pressure control.

From a previous report, concerning pump back pressure control system, criteria for maximum opening and closing rates of all valves may be established. Slow closing of valves (no water hammer) is defined as

$$t > \frac{2L}{C}$$

where t = time for a pressure wave to travel twice the duct length

L = duct length

C = velocity of sound in fluid

For the line pressure control

$$t = \frac{2L}{C} \quad \frac{(2)(565)}{3750} = 0.30 \text{ sec.}$$

To provide a reasonable degree of safety, this figure will be increased approximately by a factor of 8, which yields the maximum rate of full valve travel for the line pressure control valve equal to 2.4 seconds.

(3) Turbine Back Pressure Segregation Valve (2)--These valves shall be of the ball type for minimum pressure drop, 12 inches in diameter, and contain necessary control provisions to allow remote opening and closing. This valve is not a modulating valve. Since $t > 2L/C$, the maximum rate for full valve travel is given conservatively as

$$t \geq \frac{16L}{C} \approx 1/2 \text{ second.}$$

Additional Elements Necessary for the Ultimate Recovery System

This system requires two additional segregation valves and a turbine inlet valve capable of functioning as a turbine back pressure control during steady state pump testing. Sizes are given as follows:

(1) Segregation Valves (2)--12-inch diameter; same specifications as the turbine back pressure segregation valves.

(2) Turbine Inlet Valve (1)--12-inch continuously modulating integral control valve with very high C_V or low pressure drop. Since this valve is to maintain turbine back pressuring during

steady state pump testing the time for full valve travel may be conservatively equal to 2 seconds.

Additional Elements Necessary for the Standard Recovery System

One segregation valve upstream of the turbine and a turbine bypass valve are needed for this system. Specifications are:

(1) Turbine Inlet Segregation Valve (1)--12-inch diameter; same specification as the turbine back pressure segregation valves.

(2) Turbine Bypass Valve (1)--12-inch diameter, variable position, hydraulically actuated valve. The time for full valve travel for this valve will be 2 seconds.

In summary, control components are:

Efficiency Schedule (1 Required)

Solid state device with electrical signal proportional to $\sqrt{\Delta P_t}$.

Speed Sensor (1)

Solid state device with electrical output signal proportional to turbine speed.

Load Control Apparatus (2)

Air compressor load is controlled by a 14 inch integral control valve, where

$$K_c = .23 \frac{Q_D \text{ ft}^2}{N_D^2 \phi(A_c) \text{rpm sec.}}$$

and paralleling a 22-inch diameter sharp edged orifice. Slue rate is such that it strokes full valve travel in 3 or 4 seconds.

Water brake load is controlled by an appropriately sized integral control valve, where

$$K_c = 5.2 \frac{Q_D \text{ ft}^2}{N_D^2 \phi(A_b) \text{ rpm sec.}}$$

Slue rate is for 3 or 4-second full valve travel. Load control valves for both methods shall be spring and pressure loaded so as to increase the load when de-energized.

Segregation Valves (3) or (4)

12-inch full open or closed ball valves which will be actuated hydraulically from a remote location. Any convenient slue rate greater than 1/2 second for full valve travel is satisfactory.

Turbine Bypass Valve (1)

Variable position valve, 12 inches in diameter, hydraulically actuated from remote locations. Two seconds for full valve travel.

Turbine Inlet (Pump Back Pressure Control) Valve (1)

A 12-inch, continuously modulating integral control valve with very low pressure drop. The time for full valve travel may be in the order of 2 seconds.

Line Pressure Control (1)

A 12-inch continuously modulating integral control valve paralleling a 5.4-inch diameter sharp edged orifice. The valve slue rate should be such that full valve travel occurs in not less than 2 seconds.

VII COST ANALYSIS

Due to the limited scope of this study, it was not possible to prepare detailed specifications from which cost estimates could be made. The primary emphasis was on the major cost items, such as the turbine and power absorber, and only a budgetary type effort was considered necessary for the lesser items such as control valves and installation. While this approach simplified the overall procedure and made possible the solicitation of preliminary design type information from other companies having experience and an interest in this field, the results are necessarily varied and incomplete with respect to certain details.

Cost estimates were solicited from other companies in three general categories, control valves, installation, and heat exchanger. The response was generally negative with respect to the heat exchanger, with only two responses out of nine inquiries being obtained. Three companies out of five responded with estimates on the installation costs and one complete proposal was received for a control system.

Due to the general nature of the problem statements, there are wide variations in the interpretation of certain details which are reflected in each estimate. A typical example is the wide diversity of opinion, or experience, with respect to the cost of large scale cryogenic valves. Company C estimates a cost of $9000 +, Company A estimates $12, 000, whereas Company F proposes actual hardware at $35, 000 for a particular unit. (See Appendix E for details)

Part of this divergence is due to Company C estimates being "bare" cost, to which the various contingencies and development costs must be added. On the other hand Company F has furnished hardware for the Aerojet installation and has the advantage of experience in the assessment of cost factors. However, these costs, compared to the overall system, were generally minor, so that the spread, or range of values are not as great as the individual items may indicate.

Since the cost of the turbine and power absorber is the major part of the system cost, a careful and detailed estimate was prepared for this unit.

The breakdown is as follows:

Turbine, each	$ 215,000	
Nonrecurring charges	359,000	
Compressor, each	69,500	
Nonrecurring charges	107,000	
Cost of 2 Turbines and 3 Compressors		$ 1,104,500
Control System		
Engineering	100,800	
Hardware	141,200 (Hammel-Dahl Corp.)	
Total	242,000	
Total Cost of Turbine - Air Compressor System (without installation)		1,346,500
Heat Exchanger	340,000 to 697,000	
(Includes nonrecurring charges)		
Control System	65,000	
Total Heat Exchanger System		$ 405,000 to 762,000

Installation

The following table summarizes the cost estimates that were conducted by various contributors on the installation of the turbine and heat exchanger systems in the Aerojet Facility. Columns A through D summarize the information which was received from the various sources, and which is reproduced in the Appendix E. Column E is a composite estimate which was prepared from the inputs of A through D by AiResearch, and does not include control valves, since this item was estimated separately above.

Installation	Estimate				
	A	B	C	D	E
1. Heat Exchanger	288,000 (1)	300,000 (2)	189,000 (6)	539,000 (3)	230,000 (8)
2. Turbine and Water Brake	224,000 (4)	300,000 (2)	170,000 (7)	752,000 (5)	190,000 (8)
3. Turbine and Air Compressor	(165,000)	-	125,000 (7)	-	150,000 (8)

Notes (1) Cryogenic Control Valve cost is indeterminate

(2) Cryogenic Control Valve cost based upon Hammel Dahl quotation

(3) Cryogenic Control Valves: $274,000

(4) Cryogenic Control Valves: about $35,000 from preliminary notes

(5) Cryogenic Control Valves: $208,000

(6) Cryogenic Control Valves: $45,000

(7) Cryogenic Control Valves: $33,000

(8) Cryogenic Control Valves: Not included

Adding the estimates of Column E to the previous Totals, the final system estimates are as follows:

System	Total Cost
(1) Heat Exchanger	$ 872,000
(2) Turbine-Air Compressor	1,496,500
(3) Turbine-Water Brake	1,386,000

VIII RELIABILITY AND SAFETY CONSIDERATIONS

In considering the reliability of the various systems shown in Figure 1, the simplest recovery system (Figure 1A) constitutes a convenient reference both for efficiency of recovery and reliability. Each significant increase in recovery system efficiency is accompanied by an inevitable increase in complexity. In order to avoid a significant degradation in reliability, very conservative design practice has been adopted for the recommended system. Indeed, provision has been incorporated for some very improbable failures. For instance, runaway of the low speed turbine which might result from partial unloading due to failure of the shaft to either air compressor has been made impossible. The still operable air compressor will, at approximately 124 percent of normal rated speed, provide full power adsorbtion capability. Additionally, an automatically actuated bypass valve is incorporated into the system to allow continued running even though something disabling occured to the rotating machinery. The means for fast actuation of this valve are a hydraulic pressured (3,000 psi) line and emergency back-up accumulator. In addition, should hydraulic line or accumulator actuating pressure be lost for any reason whatever, pressure switches will indicate this fact by energizing red warning lights. Thus, it is apparent that the combination of low speed power generator (turbine), dual load adsorbers (air compressors), and stand-by bypass valve combine to instill confidence in a system of recovery more complex than the simple recovery system but whose reliability, once installed, will tend to be primarily that of the system of control which is adopted. This importance of controls is to be expected, and thus the control system warrants at least some additional investigation, with a view to self-test or check-out features which will be capable of detecting incipient troubles. From previous experience, it is known that controls and auxiliaries contribute well over 90 percent of all causes for unscheduled shutdown of large rotating machinery power installations. It was the greater number of controls and auxiliaries for the water brake system which initially led AiResearch to explore the feasibility of air braking. Air braking totally avoided the additional complication associated with a closed loop water storage and supply system.

A distinction should be made between the probability of successive repetition of desired performance, and the probability of initial achievement of such performance. The first is termed reliability and the second might be termed development prospects, the complement of which is development risk. Good system design reflects first, an acute awareneness of the importance of reliability, and second, the importance of reasonable development risk. Implementation of these precepts for the recommended system dictated (1), a low speed, balanced rotor turbine incorporating low pressure seals and possessing zero capability for runaway; (2) dual air compressor power adsorbers; and (3) an automatic fast-operating hydrogen bypass valve with dual sources (hydraulic line and accumulator) of actuation. These considerations also dictated incorporation of ball bearings in the power absorbers (thus avoiding the additional complication of an oil pressure system), and air rather than water power absorption. It is believed that all of these factors contribute to a minimum of development risk, most of which appears, at least at the time of writing, to be associated with bearings and controls.

A conventional method of providing a high reliability in continuously operating systems is the use of parallel circuits and redundant components. For an intermittant operation with long shutdown periods, essentially the same effect is obtained by simply providing spares which can be readily installed in the event of failure. Therefore, a spare turbine and compressor is recommended for this purpose. This procedure assures a high level of availability of the system.

Availability (dimensionless) is defined as the fraction of total calendar time that an installation or system is in an operationally-ready state. In order to achieve high availability, two requisites must be fulfilled: (1) a high mean time between failures or other unscheduled shut-downs, and (2) a low time for both scheduled and unscheduled maintenance action. The importance of these becomes apparent from the simple expression for availability

$$A = \frac{\text{In-readiness time}}{\text{In-readiness time + down time (all causes)}}$$

A reasonable design objective for availability of the recommended system should be a minimum of 0.99. To achieve or surpass this value, down time from all causes, including unavailability of essential replacement parts, must be limited.

REFERENCES

1. Stewart, W. L., Analytical Investigation of Single-Stage Turbine Efficiency Characteristics in Terms of Work and Speed Requirements, NACA RME56G31.

2. Miser, J., Stewart, W., and Whitney, W., Analysis of Turbo-machine Viscous Losses Affected by Changes in Blade Geometry, NACA RME56F21.

3. Cambell, C., Performance of Basic XJ79-GE-1 Turbojet Engine and Its Components, NACA RME58C12.

4. Timmerhaus, K. D. Editor, Advances in Cryogenic Engineering, Volume 8, Plenum Press, 1963.

APPENDIX A

HEAT EXCHANGER RECOVERY

Referring to the pressure-enthalpy diagram of Figure 2 of the main body of the report, the enthalpy of the high-pressure fluid is h at a pressure of P_P. The pressure drop through an upstream flow control valve and through the heat exchanger reduces the pressure to P_{HX}. The heat exchanger with the tank vapor reduces the enthalpy from h to h'. From the back pressure control valve, the pressure is again reduced to the tank operating pressure, P_1', where the fluid flashes into vapor and liquid, which is indicated by m'. After shutoff of the inlet fluid, the liquid in the tank boils until the final tank pressure P_T is attained, which results in a final liquid recovery, m. This recovery is expressed

$$m = 1 - \frac{h' - h_f}{h_{fg}} \tag{1}$$

but

$$h' = h - \Delta h \tag{2}$$

$$\Delta h = (1 - m') E C_p (T - T_f')$$

$$= (1 - m') E \Delta h_v' \tag{3}$$

$$1 - m = \frac{h - h_f'}{h_{fg}' + E \Delta h_v'} \tag{4}$$

$$\Delta h = \frac{h - h_f'}{h_{fg} + E \Delta h_v'} \tag{5}$$

$$h' = h - \frac{h - h_f'}{1 + \frac{h_{fg}'}{E \Delta h_v'}}$$

and

$$m = 1 - \frac{h - h_f}{h_{fg}} + \frac{h - h_f'}{h_{fg} (1 + h_{fg}'/E \Delta h_v')} \tag{6}$$

Since the recovery of the simple throttling system

$$m_o = 1 - \frac{h - h_f}{h_{fg}} \tag{7}$$

the net gain in recovery is

$$m - m_o = \frac{h - h_f'}{h_{fg}\,(1 + h_{fg}'/E\,\Delta\,h_v')} \tag{8}$$

where the prime notation indicates the fluid property values at operating pressures and the normal notation indicates the fluid property values at the final "use" pressure of the tank.

Figures A-1 and A-2 are the enthalpy - entropy diagrams for parahydrogen reproduced from NBS data. Specific recovery values for 15, 30 and 50 psia pressures as obtained from Equation (7), were added as shown.

Effect of Liquid Entrainment on Recovery

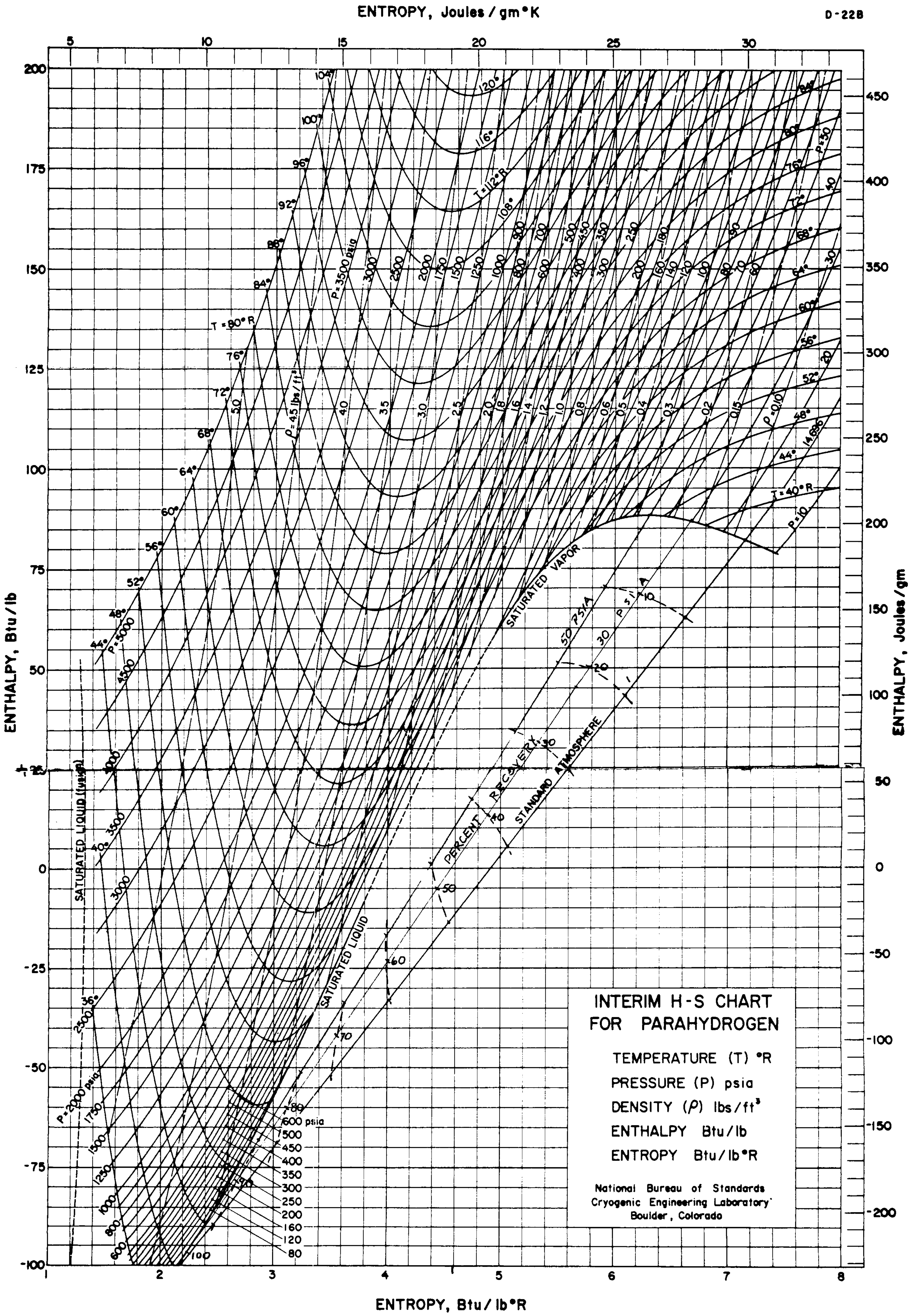
When the vent gas contains entrained liquid droplets, the primary mode of energy exchange in the inlet portion of the heat exchanger is the evaporation of the liquid droplets. Although the vapor will be heated, generally the latent heat transfer will precede the sensible heat transfer so that the liquid will be evaporated before superheating can take place. The effect of this process on recovery can readily be seen from an energy balance based upon the above sequency of heat transfer modes. It is assumed that sufficient surface area is available to provide both latent and sensible heat transfer. Then it follows that the recovery of a heat exchanger system with and without liquid entrainment can be made equal by maintaining the effectiveness.

Figure A-3 shows a diagram of the heat exchanger flow paths and the temperature profile that would exist as a result of the evaporation of entrained liquid. The corresponding energy balances may then be written:

$$\dot{w}\,c_p\,(T_{in} - T_{out}) = \dot{w}_L h_{fg} + (\dot{w}_L + \dot{w}_V)\,c_{pv}\,(T_{v_{out}} - T_{v_{in}})$$

$$\dot{w}\,\Delta h = \dot{w}_L\,h_{fg} + (\dot{w}_L + \dot{w}_V)\,E\,\Delta\,h_v$$

$$\Delta h = \frac{w_L}{\dot{w}}\,h_{fg} + \frac{\dot{w}_L + \dot{w}_V}{\dot{w}}\,E\,\Delta\,h_v$$



The following charts for parahydrogen are available in 17" x 22" size from the Cryogenic Data Center, National Bureau of Standards, Boulder, Colorado:

In Metric Units				In British Units			
D-20A	T-S Chart	20 to 100°K	1 to 340 atm.	D-20B	T-S Chart	36 to 180°R	10 to 5000 psia.
D-21A	T-S Chart	80 to 300°K	1 to 100 atm.	D-21B	T-S Chart	140 to 540°R	10 to 1500 psia.
D-22A	H-S Chart	20 to 60°K	1 to 340 atm.	D-22B	H-S Chart	36 to 100°R	10 to 5000 psia.

Prepared for: National Bureau of Standards, Technical Note, TN 130 (PB161631) December 1961, "Provisional Thermodynamic Functions for Parahydrogen", H. M. Roder and R. D. Goodwin; by the Cryogenic Data Center, National Bureau of Standards, Boulder, Colorado from property functions reported in NBS TN 130. These functions were used to calculate temperature and entropy for all intersections of isobars and isenthalps and for intersections of isobars and isometric lines. Additional points were also calculated as necessary to complete the precise definition of the property lines.

R. B. Stewart, R. D. McCarty, L. J. Ericks (December 1961)

Figure A-1. Properties of Parahydrogen

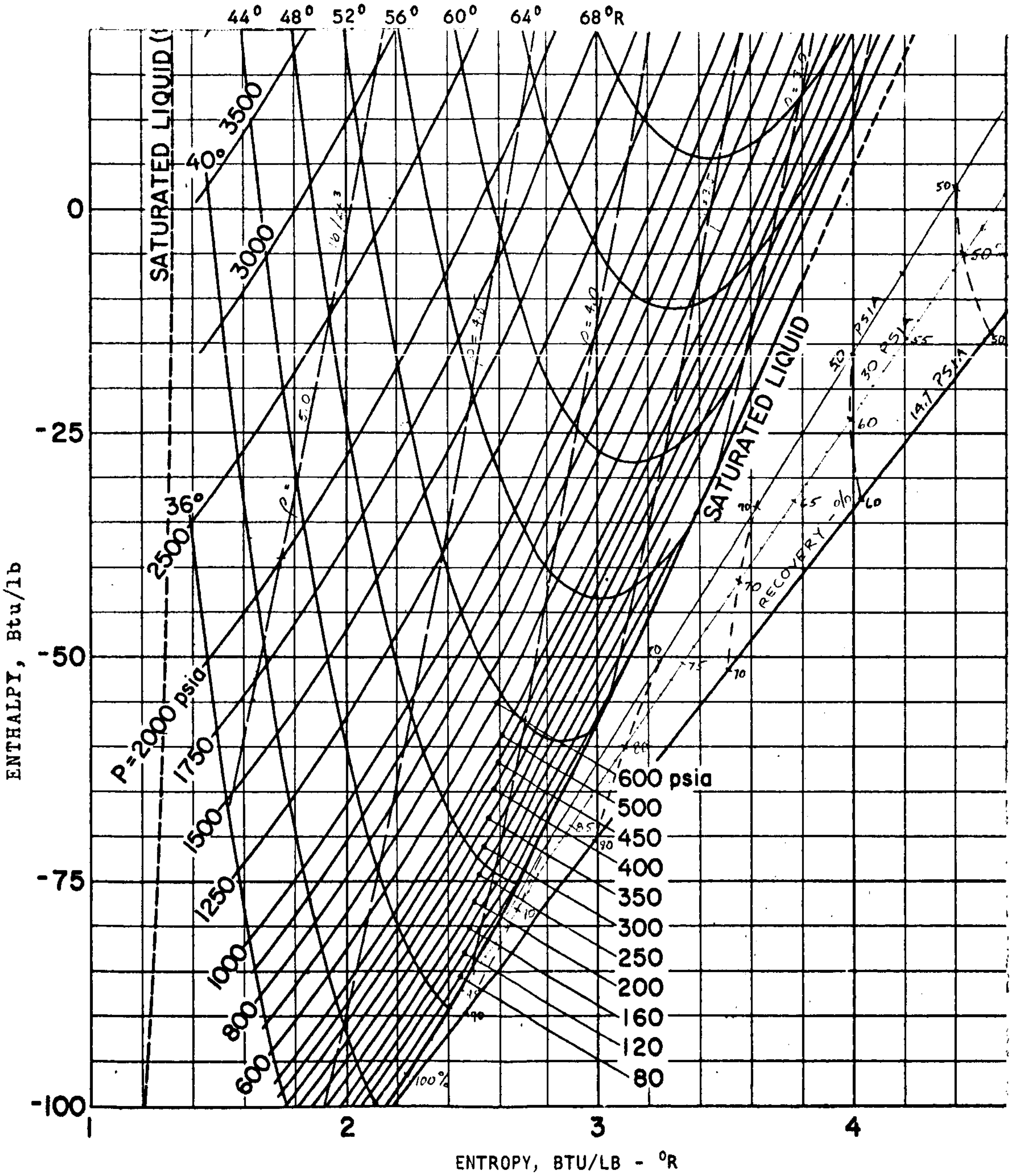


Figure A-2. Properties of Parahydrogen

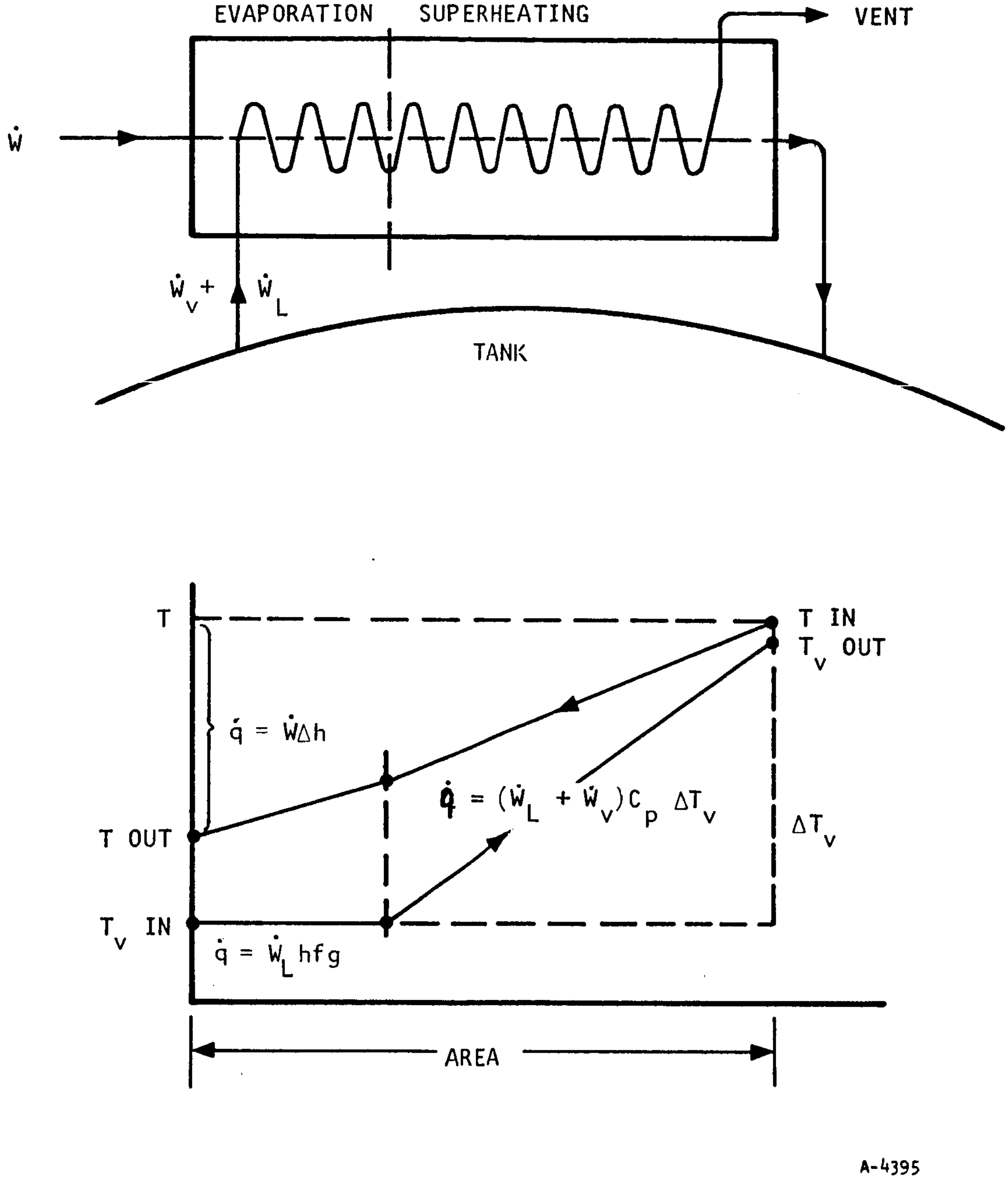


Figure A-3. Diagram of Heat Exchanger Performance with Liquid Entrainment in Vent Gas where $\dot{w}_L$ = liquid flow rate

$\dot{w}_V$ = vapor flow rate

$\dot{w}$ = total flow rate

But the specific recovery, $m = 1 - \dfrac{\dot{w}_L + \dot{w}_V}{\dot{w}}$

and, $$1 - m = \frac{h' - h_o}{h_{fg}} = \frac{\dot{w}_V}{\dot{w}}$$

Then $$(1 - m)\, E\, \Delta h_v = h - h' - \frac{\dot{w}_L}{\dot{w}} h_{fg}$$

$$= h - \left(h_o + h_{fg}\left[1 - m - \frac{\dot{w}_L}{\dot{w}}\right]\right) - \frac{\dot{w}_L}{\dot{w}}\, h_{fg}$$

$$= h - h_o - h_{fg}\left(1 - m - \cancel{\frac{\dot{w}_L}{\dot{w}}} + \cancel{\frac{\dot{w}_L}{\dot{w}}}\right)$$

Thus, the liquid terms cancel out and

$$m = 1 - \frac{h - h_o}{h_{fg} + E\,\Delta h_v}$$

which is the same as the recovery of the heat exchanger system without entrainment. However, from Figure A-3, it is apparent that additional area, or conductance, is necessary to maintain the effectiveness E.

APPENDIX B

Heat Leakage and Cooldown Relations

The operation of equipment at cryogenic temperatures involves the expenditure of relatively high grade forms of energy for the attainment and maintenance of operating temperature. For liquified gases such as hydrogen and helium, elaborate refrigeration devices or cryostats are required for the removal of energy from the fluid and from the associated equipment. This cost is partially reflected in the cost of the fluid since an excess of fluid over the direct operational requirements is generally used to obtain and maintain the operating temperature. In addition, specialized equipment is needed, such as vacuum jacketed insulation, vacuum pumps, cryostats and/or liquid-vapor circuits with the associated control systems. The cost and complexity of these devices plus the additional fluid represent an important aspect of the overall system design.

For an individual component, such as a tank or a pump, the cooldown and heat leakage requirements can be stated in relatively simple terms. The amount of cryogenic liquid is proportional to the heat capacity of the equipment divided by the heat capacity of the liquid, both latent and sensible. The maximum requirement is when only the latent heat of the liquid is used, as exemplified by immersion in a liquid bath. Here the boil-off vapor escapes without further contact with the equipment. The minimum requirement is obtained when the vapor is also utilized for cooling. When the escaping vapor temperature approaches that of the equipment, the liquid requirement is a minimum. An example would be a long pipe into which a small amount of liquid is admitted in one end, and the pipe and vapor temperature are nearly equal at the other, as the cooldown progresses through the length of the pipe. Obviously, the rate of cooldown is an important parameter since any geometry can achieve a maximum sensible heat exchange if equilibrium temperature are attained before release of the vapor.

In the above discussion, only the heat exchange between the fluid and component was considered, as may occur in a perfectly insulated enclosure. Actually, heat leakage through an insulation will require an additional boil-off of fluid which is proportional to the outside surface area, the effective conductance of the insulation, and the time duration of the low temperature. Although there is a similar maximum and minimum liquid requirement as for cooldown, this leakage is generally associated with low flow conditions in which the sensible heat of the vapor is also effective. Therefore the amount of liquid required for heat leakage is generally the minimum indicated by the latent and sensible heat capacities.

Although the cooldown and heat leakage modes of heat transfer occur simultaneously, the rate of cooldown is generally high so that heat leakage effects can be ignored during cooldown. Then each mode can be considered as successive occurrances in an overall procedure. Thus a tank which is used infrequently or for a short time period may undergo a limited number of cooldowns rather than be maintained at a low temperature. For other components and conditions, the cooldown requirements may be excessive and a continuous boiloff to maintain operating temperatures may be preferable.

Clearly a combination of the two modes, that is, the maintenance of an intermediate low temperature and then cooldown for each operation can provide an optimum approach, with respect to the cost of insulation and the cost of liquid boil-off. An intermediate temperature reduces the heat leakage during the standby period and reduces the cooldown fluid for each run. In this approach, the intermediate temperature generally is dependent upon the number of cooldowns versus the duration and heat leakage between runs, and for an optimum design, should utilize the sensible heat of the fluid such that the vapor discharge temperature approaches the maximum equipment temperature for the major portion of the boil-off flow rate.

It may be convenient in the design of cryogenic equipment to determine a "boil-off" penalty similar to the drag and weight penalties associated with airborne equipment. This penalty would involve not only the cost of boil-off fluid but also the cost of insulation, which, in turn would be dependent upon equipment mass, material specific heats, surface areas, and insulation conductances. Thus it is evident that size, weight, and shape, which are normally not critical in ground installations, may become very important parameters in cryogenic systems.

The following sections present the basic thermodynamic relations from which design and optimization parameters can be obtained.

1. Component Cooldown

The maximum liquid required for cooldown from normal to liquid temperature may be expressed:

$$W_\beta = \frac{W_e \int_{T_o}^{T_a} C_e \, dT_e}{\lambda} \tag{1}$$

where C_e = Specific heat of the equipment material

W_e = Weight of the material

T_a = Initial temperature of material, ambient

T_e = Material temperature

T_o = Final or liquid temperature

λ = Heat of vaporization of the liquid

For the minimum cooldown, the liquid requirement is,

$$W = W_e \int_{T_e}^{T_a} \frac{C_e \, dT_e}{\lambda + \int_{T_o}^{T_e} Cp \, dT_v} \tag{2}$$

where Cp = Specific heat of the vapor

T_v = vapor temperature

These integrals have been evaluated for various materials and cryogenic fluids in Reference 4. The curves for cooldown of copper, steel, and aluminum in liquid hydrogen are reproduced in Figure B-1, from which the amount of liquid can readily be determined for any initial and final temperature.

2. Component Heat Leakage Boiloff Requirements

The liquid boil-off which is required as a result of heat leakage from the external environment is determined from an energy balance,

$$\dot{w}_\beta = U A_1 \frac{(T_a - T_o)}{\lambda} + \frac{U A_2 \Delta T_{\ell m}}{\bar{C}_p (T_v - T_o)} \tag{3}$$

where $\dot{w}_\beta$ = flow rate of liquid

A_1 = area where boiling occurs

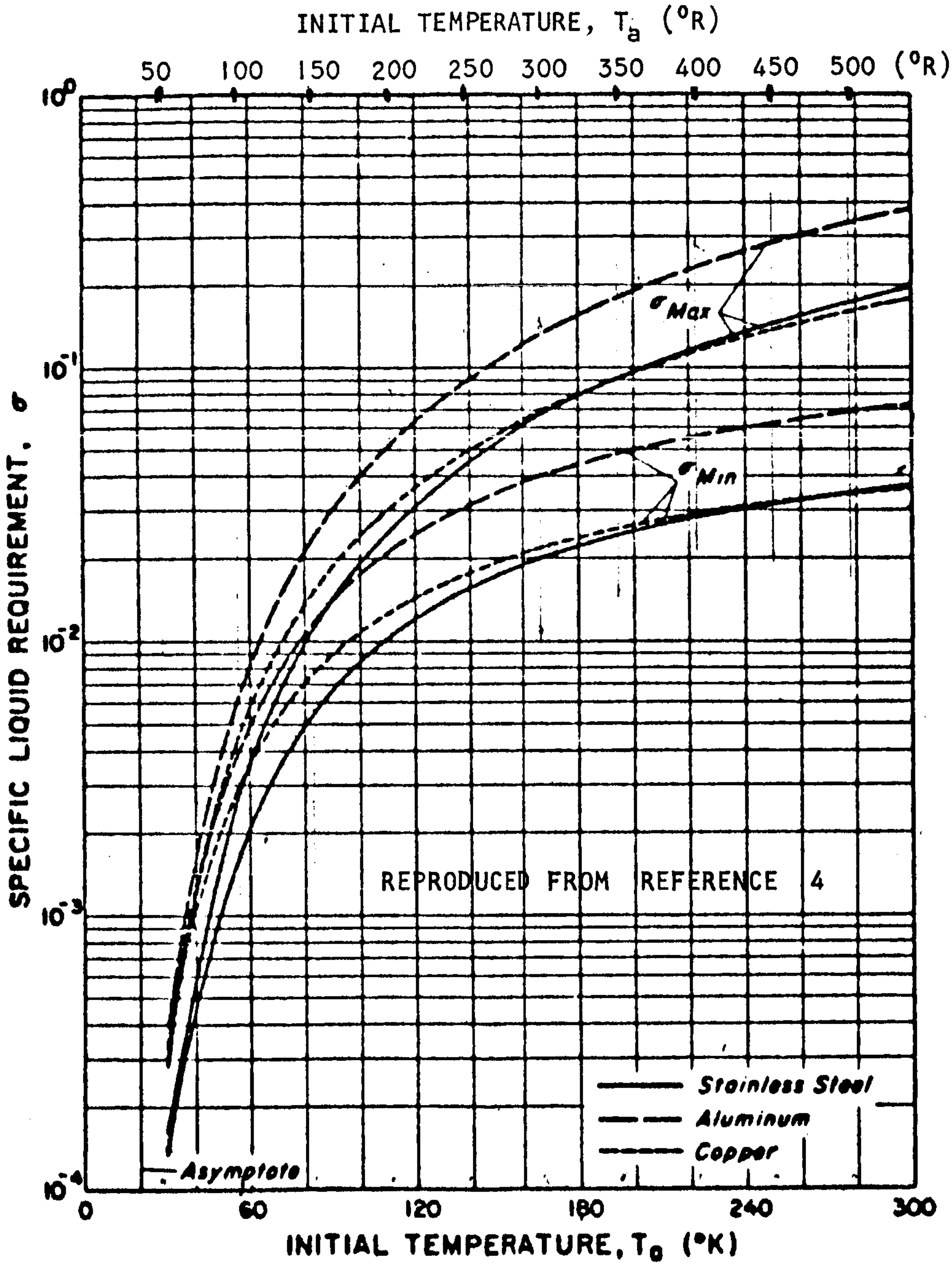


Figure B-1. Cooldown Requirement for Materials in Liquid Hydrogen

A_2 = area of sensible heat exchange

$\overline{C}_p$ = average specific heat of vapor

T_a = external temperature

T_v = vapor temperature at exit

T_o = fluid boiling temperature

ΔT_{lm} = log mean temperature difference between T_a and T_v

U = conductivity across insulation

In this expression, the first term on the right indicates the amount of boil-off and the second term indicates the temperature rise of the resulting vapor.

Since
$$A_1 = \frac{\dot{w}\,\lambda}{U\,(T_a - T_o)}$$

$$A_2 = \frac{\dot{w}_\beta\, C_p\,(T_v - T_o)}{U\,\Delta T_{\ell m}}$$

the liquid flow rate required for a given area and heat conductance is

$$\frac{\dot{w}_\beta}{UA} = \frac{1}{\dfrac{\lambda}{T_a - T_o} + \overline{C}_p\,\ell n \left[\dfrac{1}{1 - \dfrac{\Delta T_v}{T_a - T_o}}\right]} = \gamma \qquad (4)$$

where $\dfrac{\Delta T_v}{T_a - T_o}$ is the ratio of vapor temperature rise to the maximum temperature difference.

Using the following values for liquid hydrogen at atmospheric pressure in Equation (4),

λ = 190 Btu/lb

$$T_a - T_o = 520^\circ - 36^\circ = 484^\circ$$

and
$$\frac{\Delta T_v}{T_a - T_o} = .01 \text{ to } 1.0,$$

a curve of γ vs T_v is obtained as shown in Figure B-2. From this curve, it is evident that the maximum rate is $\frac{T_a - T_o}{\lambda}$ as obtained without superheat of the vapor. The minimum value, zero boil-off, results from the infinite area, A_2, required to obtain $T_v = T_a$, which is not a practical consideration.

Minimum Boil-off

As discussed previously, the optimum temperature level is obtained from the combination of the boil-off fluid required to maintain a partial cooldown and the number of cooldowns to be made in a given period of time. Thus the total liquid requirements may be obtained by adding Equations (2) and (4),

$$\frac{W}{\beta} = \left[NW_e \int_{T_v}^{T_u} \frac{C_e \, dT_e}{\lambda + \int_{T_o}^{T_v} C_p \, dT_v} \right] +$$

$$\left[\frac{UA \; \Delta\theta}{\frac{\lambda}{T_a - T_o} + \overline{C}_p \, \ell n \left(\frac{1}{1 - \frac{\Delta T_v}{T_a - T_o}} \right)} \right]$$

where $\Delta \theta$ = time duration between cooldowns

N = number of cooldowns

U = conductance across insulation

= k/x, approximately, for cryogenic insulation

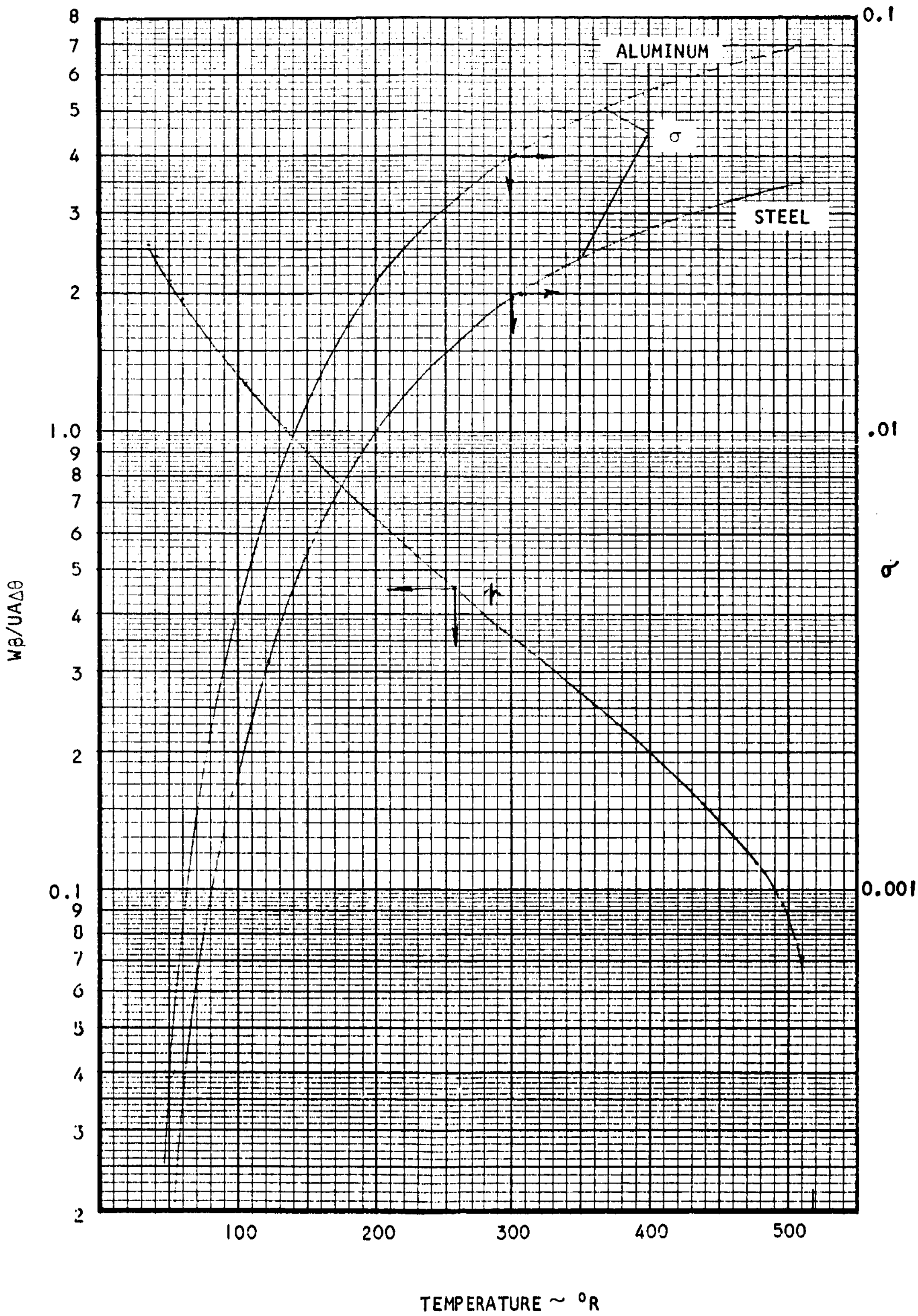


Figure B-2. Cooldown Parameters vs Temperature k = effective conductivity of insulation

x = thickness of insulation

When this expression is simplified,

$$\underline{W}_{\beta} = NW_e\sigma + UA_s \, \Delta \theta \, \gamma$$

the physical, operational, and temperature dependent variables are more clearly defined. Here N and $\Delta\theta$ are the independent variables, which are determined by the operational characteristics of the system. W_e and UA_s are functions of the physical characteristics of the equipment, and σ and γ govern the effects of the temperature level on the boil-off requirement. The first (cooldown) term on the right increases with the temperature, whereas the second (heat exchanger) term decreases, which means that an optimum or minimum flow exists at some intermediate temperature. For a number of cooldowns, N, over a period of time, $\Delta\ \theta$, the optimum boil-off is dependent upon the quality of the insulation, U, and the ratio of weight to surface area, W_e/A_s. Therefore, the optimum temperature may vary for different components, such as piping, pump or turbine, heat exchanger, tank, or valves. In general, the ratio of weight-to-surface-area is a maximum for valves, turbomachinery, and heat exchangers, and is a minimum for piping, which implies that the insulation may be varied to obtain the same optimum temperature level.

When the physical terms are lumped into a single parameter (UA/W_e), two expressions may be obtained:

$$\frac{\underline{W}_{\beta}}{UA\,\Delta\theta} = \frac{N}{\Delta\ \theta}\left(\frac{W_e}{UA}\right)\sigma + \gamma$$

or

$$\frac{\underline{W}_{\beta}}{NW_e} = \sigma + \left(\frac{UA}{W_e}\right)\left(\frac{\Delta\theta}{N}\right)\gamma$$

Either expression shows the dependency of the boil-off on the conductance-area/weight ratio. The possible trade-offs between temperature level and insulation conductance for minimizing the boil-off fluid required are also indicated by this parameter.

As an example, consider a section of piping, made of steel, having a wall thickness of 1/2-in. with vacuum-jacketed insulation. The weight-area ratio is

$$\frac{W_e}{A_s} = \rho\ t = 500/24 = 21$$

For the heat exchanger described in a previous section, the corresponding ratio is 50,000/1000 = 50.

These values indicate that the effects of heat transfer will be greater for the piping than for the heat exchanger and that a higher maintenance temperature can be used for minimum boil-off. However, the optimum level must be determined for each component for a variety of insulation thicknesses to determine the overall boil-off requirements and resulting optimum temperature level. Using a pipeline 350 ft long and a 2-in. vacuum-jacketed insulation (k/x = .008 Btu/hr-ft^2-$^\circ$F) for both the line and the heat exchanger, the required liquid boil-off is obtained as shown in Figure B-3.

These results show that each component has an optimum temperature level which may differ considerably from the optimum level for the entire system. By calculating various curves for each component and combining them into a system requirement curve, the relative importance of individual insulations can be readily determined, as shown by the dotted lines in the figure.

Transient State Analysis

An approximate transient state analysis can be performed by considering the internal masses of the pipe and turbine as lumped parameters in which the temperatures at any time is evenly distributed. For short transfer lines, a minimum number of divisions would be two, one for the lines and one for the turbine, or heat exchanger. Two divisions are manageable by iterative slide rule computation; three or more divisions or "lumps" will require programming for digital computer solutions.

Although the Aerojet System should have more than two divisions for an adequate solution, this approach was considered beyond the scope of the present effort. Nevertheless, it was desireable to check out the basic equations for this section of the study, and a two lumped parameter calculation was made on the turbine and heat exchanger recovery systems. The results are given in Figures B-4a and B-4b. The basic relations used are as follows.

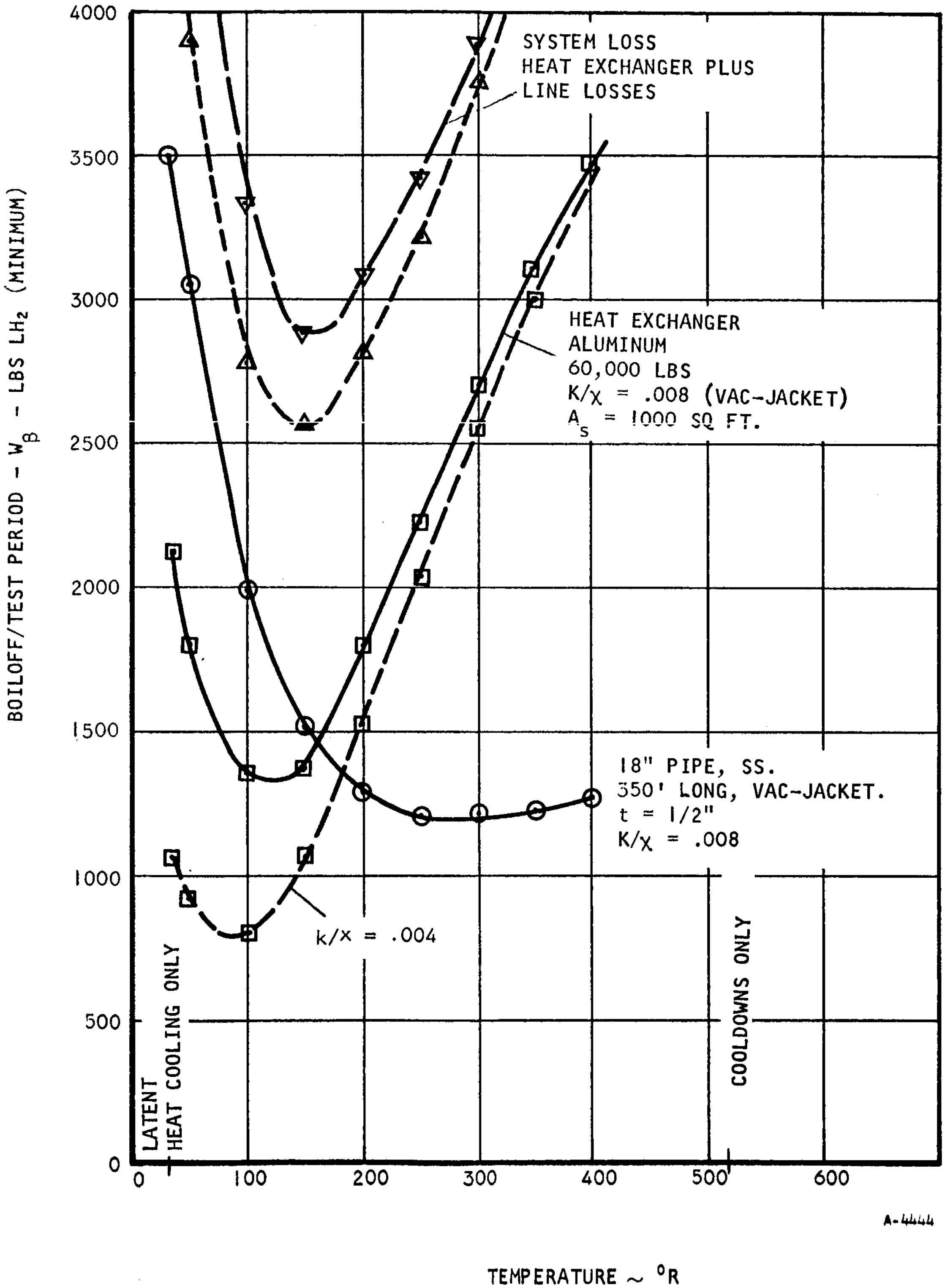


Figure B-3. Liquid Hydrogen Boiloff Requirements for Various Maintenance Temperatures

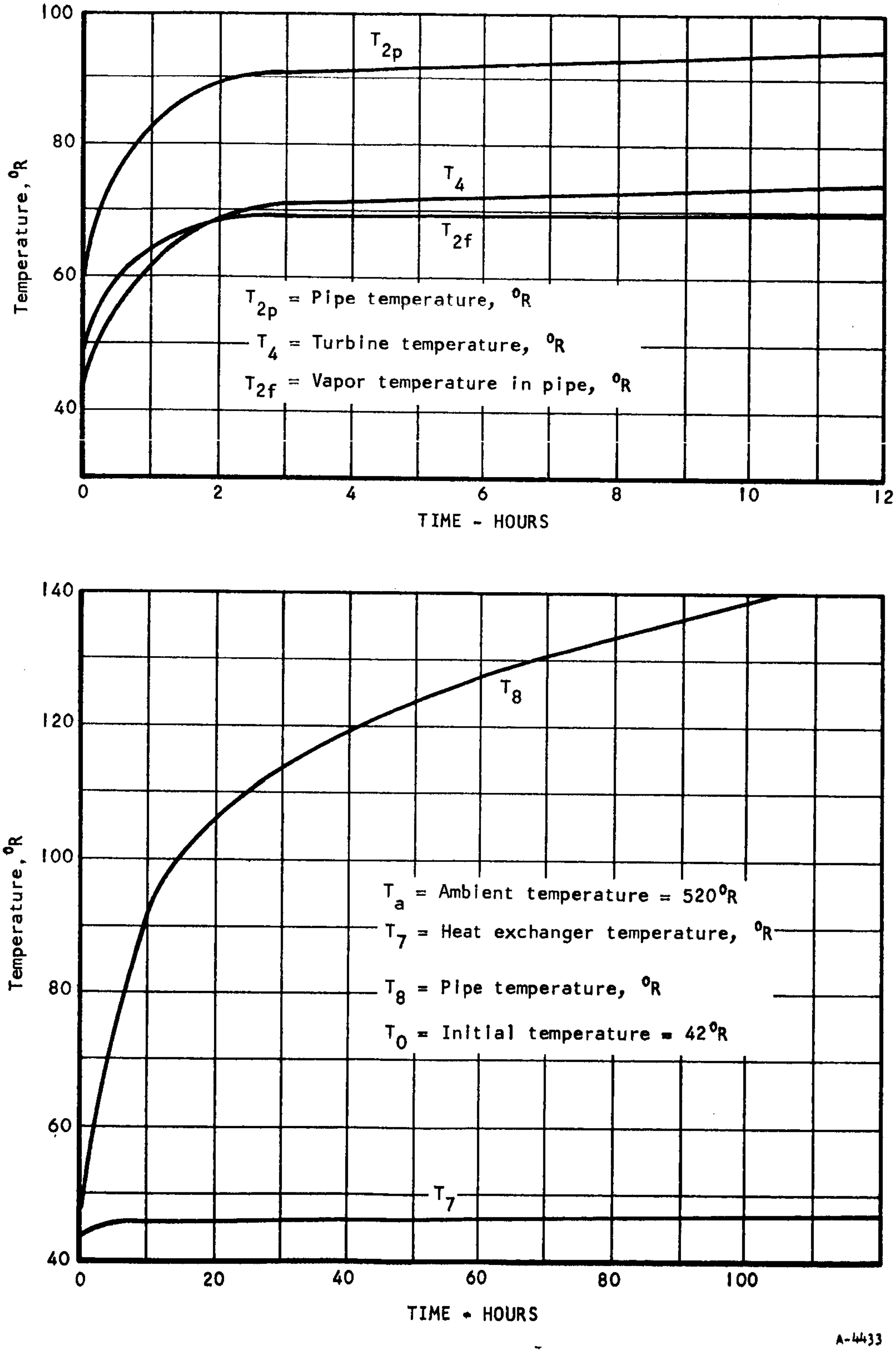
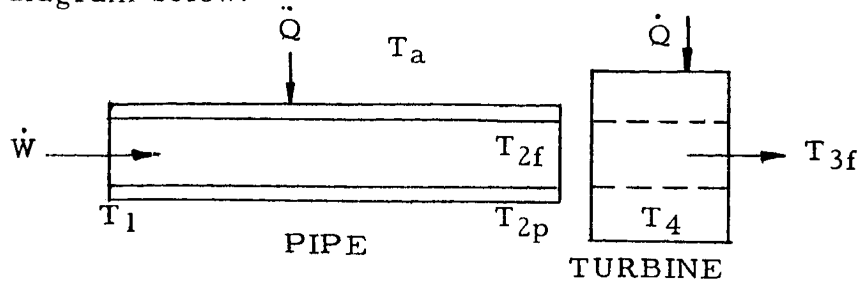


Figure B-4. Temperatures, Transient State.

Turbine System

The system may be represented as shown in the schematic diagram below:



The heat balance for Section 1-2 may be written:

$$\frac{k}{L} A_o \left[T_a - \left(\frac{T_1 + T_{2p}}{2} \right) \right] - hA_i \left(\frac{T_{2p} - T_{2f}}{2} \right) = \frac{dE}{d\theta} \qquad (1)$$

The energy stored in the pipe is:

$$E = M \left(\frac{C_1 + C_{2p}}{2} \right) \left(\frac{T_1 + T_{2p}}{2} \right) \qquad (2)$$

and

$$\frac{dE}{d\theta} = \frac{M}{4} \left(C_1 + C_{2p} \right) \left(\frac{dT_{2p}}{d\theta} \right) \qquad (3)$$

But the heat transferred from the pipe to the fluid must be equal to the heat gained by the fluid, or:

$$hA_i \left(\frac{T_{2p} - T_{2f}}{2} \right) = \dot{W} \left(\frac{C_{p1} + C_{p2}}{2} \right) \left(T_{2f} - T_1 \right) \qquad (4)$$

By solving Equation (4) for T_{2f}, and substituting for $\frac{dE}{d\theta}$ from Equation (3) into Equation (1), a pointwise solution for $\frac{dT_{2p}}{d\theta}$ may be accomplished using small changes in T_{2p}.

The heat balance for Section 2-3 can be written:

$$\frac{k}{L} A_T \left(T_a - T_4\right) - hA_h \left[T_4 - \left(\frac{T_{2f} + T_{3f}}{2}\right)\right] = \frac{dE}{d\theta} \quad (5)$$

Here the energy stored in the pipe is:

$$E = MC_4 T_4 \quad (6)$$

and,

$$\frac{dE}{d\theta} = M C_4 \frac{dT_4}{d\theta} \quad (7)$$

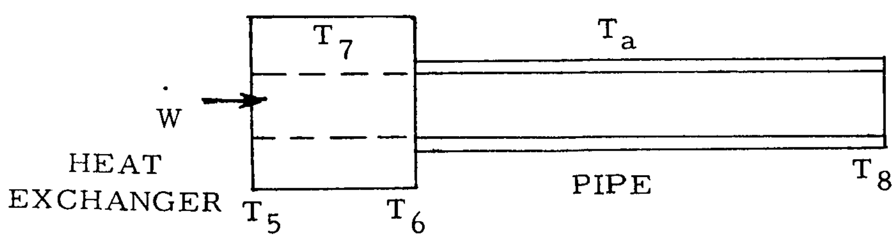
Here again, the heat transferred from the turbine to the fluid must equal the heat gained by the fluid, or:

$$hA_h \left[T_4 - \left(\frac{T_{2f} + T_{3f}}{2}\right)\right] = \dot{W} \left(\frac{C_{p2f} + C_{p3}}{2}\right)\left(T_3 - T_{2f}\right) \quad (8)$$

Again, a pointwise solution for $\frac{dT_4}{d\theta}$ may be evaluated by solving Equation (8) for T_3 and substituting in Equation (5) for T_3 and for $\frac{dE}{d\theta}$ from Equation (7).

Heat Exchanger System

The heat exchange system is represented schematically below.



The heat balance for Section 5-6 may be written:

$$\frac{k}{L} A_o \left(T_a - T_7\right) - hA_i \left[T_7 - \left(\frac{T_5 + T_6}{2}\right)\right] = \frac{dE}{d\theta} \quad (9)$$

The energy stored in the heat exchanger may be written:

$$E = M\, C_7 T_7 \tag{10}$$

and,

$$\frac{dE}{d\theta} = M\, C_7 \, \frac{dT_7}{d\theta} \tag{11}$$

The heat transferred from the heat exchanger to the fluid must equal the heat gained by the fluid.

$$hA_i \left[T_7 - \left(\frac{T_5 + T_6}{2} \right) \right] = \dot{W} \left(\frac{C_{p5} + C_{p6}}{2} \right) \left(T_6 - T_5 \right) \tag{12}$$

With Equation (12) solved for T_6 and Equation (11) solved for $\frac{dE}{d\theta}$, the substitution necessary for pointwise solution may thus be made in Equation (9).

The heat balance for Section 6-8 can be written:

$$\frac{k}{L}\, A_o \left[T_a - \left(\frac{T_6 + T_{8f}}{2} \right) \right] - hA_i \left(\frac{T_{8p} - T_{8f}}{2} \right) = \frac{dE}{d\theta} \tag{13}$$

where:

$$E = M \left(\frac{C_6 + C_8}{2} \right) \left(\frac{T_6 + T_{8p}}{2} \right) \tag{14}$$

and

$$\frac{dE}{d\theta} = \frac{M}{4} \left(C_6 + C_8 \right) \left(\frac{dT_6}{d\theta} + \frac{dT_{8p}}{d\theta} \right) \tag{15}$$

Again, all heat transferred from the pipe to the fluid must be gained by the fluid. Therefore

$$hA_i \left(\frac{T_{8p} - T_{8f}}{2} \right) = \dot{W} \left(\frac{C_{p6} + C_{p8}}{2} \right) \left(T_{8f} - T_6 \right) \tag{16}$$

Equations (16) and (15) can be solved for T_{8f} and $\frac{dE}{d\theta}$. Substition into Equation (13), permitts a pointwise solution for $\frac{dT_{8p}}{d\theta}$.

APPENDIX C

TURBOMACHINERY DESIGN STUDIES

Range of Operation Considerations

During the initial phase of the study, it became evident that off-design point considerations would have an important bearing on the turbine and power absorber design. Pump tests are generally conducted over a range of conditions to establish the performance characteristics, and this range would either be met by an appropriately designed turbine-absorber unit or would require a bypassing of fluid during the excursion tests. To maximize recovery, the first approach was to design a turbine for the full range of pump operation. This maximum range as supplied by Aerojet, is shown in Figure C-1.

1. High Speed Centrifugal Turbine Design

The following table summarizes the results of design calculations for a centrifugal turbine design having a diameter of 15 inches and a speed of 20,000 rpm. This machine is particularly well suited for a variable area nozzle design, which will provide a high efficiency over a wide range of pump operating conditions.

A cross sectional drawing is shown in Figure 17 of the main report test.

Impeller

D_1	impeller diameter	15 inches
H_{ad}	adiabatic head	68.6 Btu/lb
U_1	tangential velocity	1310 ft/sec
α_1	nozzle angle	20°
C_{m1}	radial velocity	440 ft/sec
w_1	blade velocity, inlet	450 ft/sec
w_2	2.2 x w_1, velocity, outlet	990 ft/sec

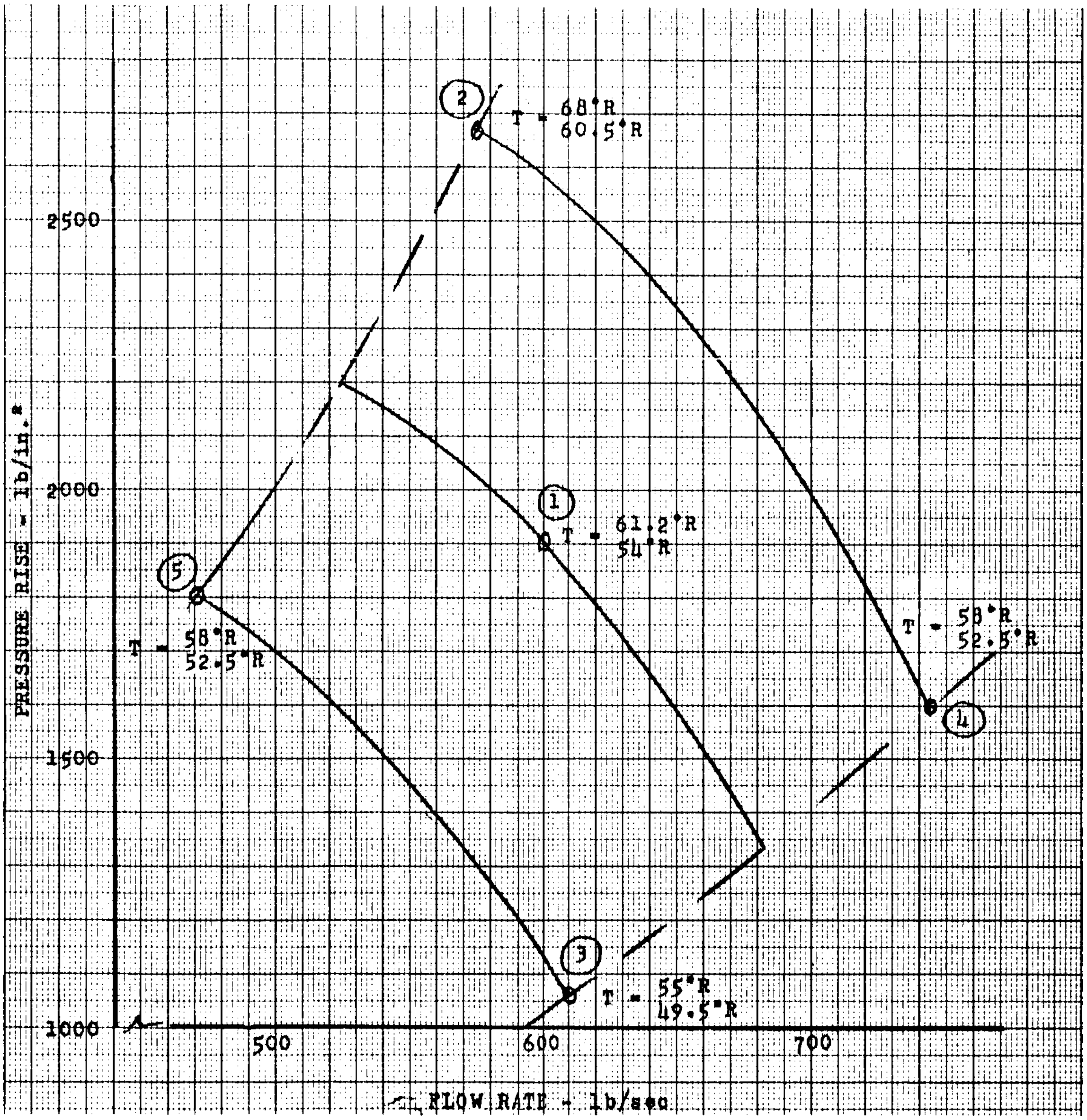


Figure C-1. LH_2 Fuel Pump Performance Range of Test Stand Operation

Impeller (continued)

β_{b-2}	blade angle, outlet, tip	30^{o}
C_2	discharge velocity	495 ft/sec
U_2	exducer tip speed	858 ft/sec
d_2	exducer diameter	9.85 in.
	$C_2^2/2g\,J$	4.9 Btu/lb
C_3	discharge line velocity	200 ft/sec
	$C_3^2/2g\,J$	0.8 Btu/lb
P_{2S}	exducer static pressure	80 psia
T_2	exducer temperature	$50^{o}R$
A_e	exducer flow area	0.32 sq ft
Ab/An	vane blockage	14 percent
d_{hub}	hub diameter	4.95 in.

Nozzle

ς	loss factor	4 percent
	$C_1^2/2g\,J$	34.2 Btu/lb
P_N	nozzle discharge pressure	920 psia
T	nozzle discharge temperature	$55^{o}R$
ρ_N	nozzle discharge density	4.25 lb/ft^3
Ac_m	wheel inlet area	0.32 ft^2
b	blade height	0.98 in.

Discharge Line

d_3	discharge line diameter	11.9 in.
q_3	discharge line velocity head	16.4 psi
P_{3s}	discharge line static pressure	163.6 psia

Performance

The variable nozzle area turbine has a high efficiency over a broad range of operating conditions. This range includes both pressure levels and flow rates. For the range indicated by the M-1 test pump, as given in Figure C-1, the following performance is anticipated from the preliminary calculations.

Parameter	Design Point 1*	High Pressure Test Point 2*	Low Pressure Test Point 3*	High Flow Test Point 4*	Low Flow Test Point 5*
w, lb/sec	588	575	650	744	470
P_o, psi	1900	2670	1070	1600	1800
T_o, ^oR	61	68	55	58	58
m_o, %	46	24.5	66.5	55	52
Δh_p, Btu/lb	69	105	39	56	66
η_t	.92	.90	.89	.87	.89
Δh_t, Btu/lb	63.5	94.0	34.6	55.5	59
N, rpm	20,000	23,200	15,000	17,700	19,400
HP	53,000	76,500	32,000	52,000	39,000
m_t-m_o, %	35	51.0	18.9	30.3	32
m_t %	81	75	85.4	85.3	84

These values are for the upper temperature levels given in Figure C-1. For the lower temperature levels, the turbine net recoveries, $(m_t - m_o)$ are smaller but the overall recovery, m, is greater. Therefore, the above table indicates the maximum potential recovery of the variable nozzle turbine system.

Axial Turbine Design

In order to arrive at the best vector diagrams for the axial staging, an optimization analysis was conducted which utilized the following definition of blading work coefficient:

λ = work coefficient = $U/\Delta V_u$

where U = blade speed

ΔV_u = total change in fluid tangential velocity

For an impulse stage, λ = 0.50

For a 50% reaction stage, λ = 1.00

An expression for stage total efficiency can be derived as follows:

$$\eta_{T-T} = \frac{\lambda}{\lambda + \frac{1}{2}\left\{K \frac{(R_{eh})^{-1/5}}{\cot \alpha_1}\left(6 \cot^2 \alpha_1 + 4\lambda(\lambda - 1) + 3\right)\right\}}$$

where α_1 = stator angle, measure from axial deg.

K = blade row loss constant of proportionality

Re_h = height Reynolds number (based on blade height, exit blade row velocity, and inlet state condition)

K and R_{eh} are defined more fully in References 1 and 2. The above efficiency derivation is based on the assumption of axial outflow from all rotors and constant axial velocity through a given stage.

Calculations were made using the foregoing expression over a range of hub work coefficients, λ_h, and hub stator angles, α_{1h}. Free vortex flow conditions were assumed for both stator and rotor. The results are plotted on Figure C-2.

Maximum efficiencies are obtained for nozzle angles between 60° and 70°, and work coefficients greater than 0. 6. This range, however, is denoted by high diameters and low blade height, and leakage corrections must then be added. For purposes of comparision, a clearance annulus of 0. 020 inch was assumed for both rotor and stator; and the efficiencies of the optimum region were readjusted as shown in Figure C-3. The hub stator angle, $\alpha_{1h} = 70°$ is clearly the optimum value with the peak efficiency occurring between $\lambda = 0.7$ and 0. 8. Since $\lambda_h = 0.7$ yields a lower overall diameter, it was selected for the final design.

Cavitation Analysis

Defining the discharge specific speed as N_d

$$N_d = \frac{N Q_3^{1/2}}{(h_{d_v})^{3/4}}$$

then,

$$N_d = \frac{1897 \left[\dfrac{Q_3}{N D_{t2}^3} \right]^{1/2}}{\left\{ 591 \left(\dfrac{1 + k}{k_b} \right) \left[\dfrac{Q_3}{N D_{t2}^3} \right] \left(\dfrac{1}{1 - \epsilon^2} \right)^2 + k \right\}^{3/4}}$$

where N = rotating speed, rpm

Q_3 = rotor discharge annulus volume flow, cfs

D_{t2} = rotor tip diameter, ft

k = diffusion parameter

k_b = blade blockage factor

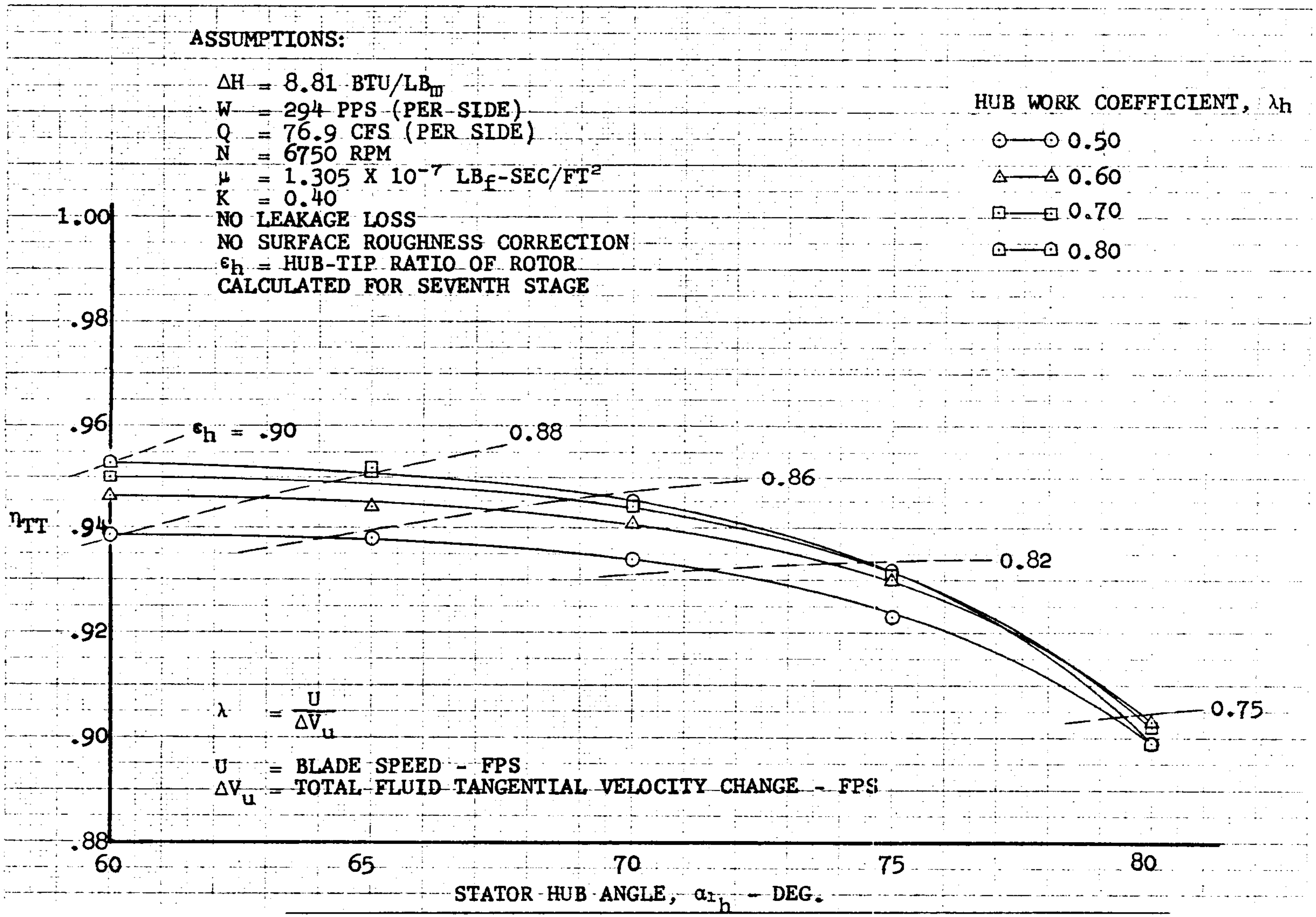


Figure C-2. Effect of Hub Stator Angle and Hub Work Coefficient on Attainable Stage Efficiency

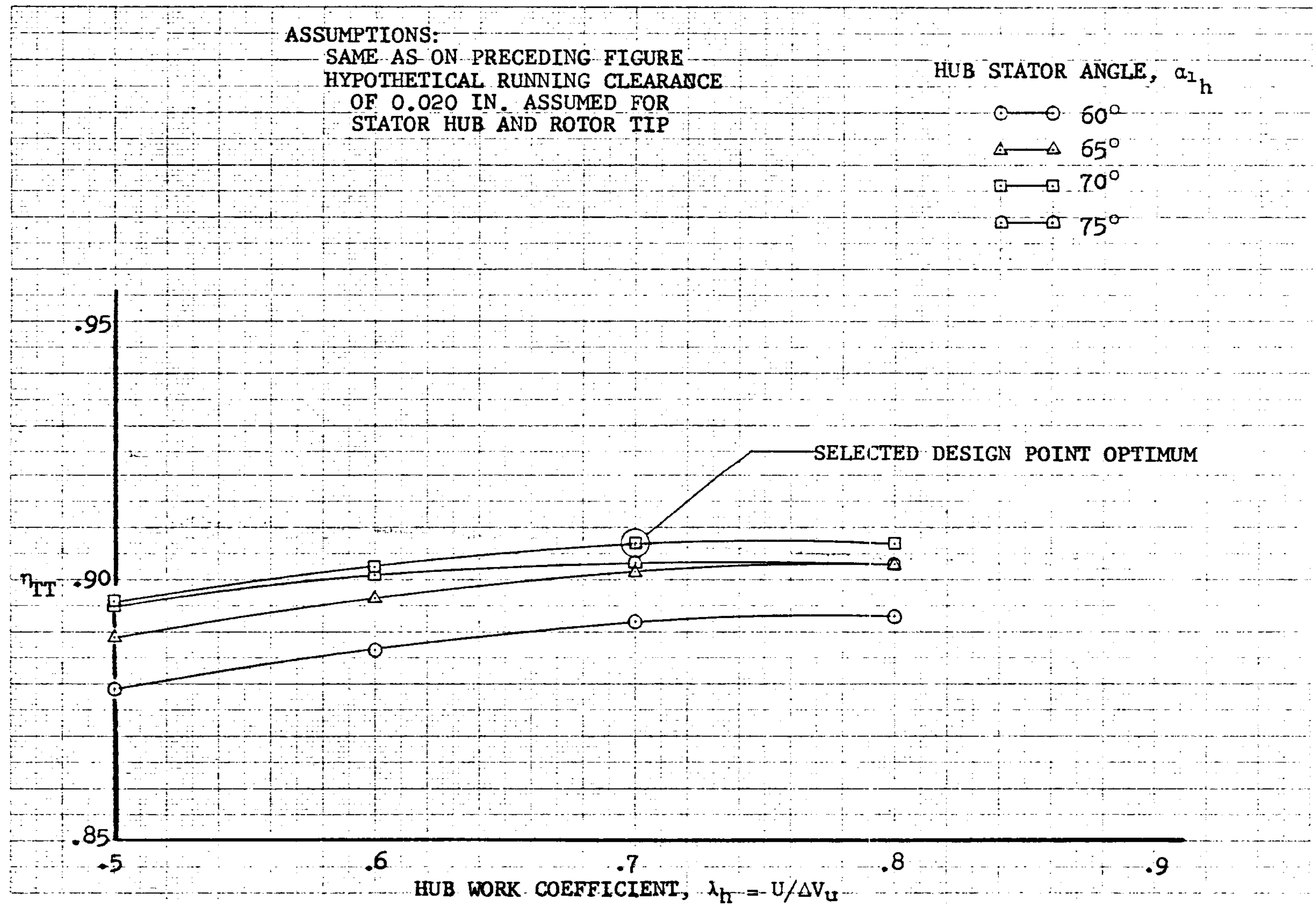


Figure C-3. Effect of Tip and Hub Leakage on Maximum Attainable Performance

ϵ = rotor hub tip ratio

h_{dv} = net positive discharge head (NPDH), ft

This relationship can be derived from energy and continuity considerations, by equating the head at the minimum pressure point (peak velocity) on the suction side of the blade at the rotor tip equal to vapor conditions.

A parameter which is important in the design of turbine blade profiles is the suction surface diffusion parameter, D_s, defined as:

$$D_s = 1 - \frac{\omega 2}{\omega 1}$$

where ω_2 = cascade exit velocity

ω_1 = peak velocity on suction surface of blade

For optimal blade performance, the blade should be designed for $D_s = 0$. For highly cambered blades, this goal is sometimes rather difficult to achieve. The factor k in the above relation can be substituted by:

$$k = \frac{D_s(2 - D_s)}{(1 - D_s)^2}$$

Calculations can now be made of N_d versus $(Q_3/N D_{t2})^3$ for a range of D_s.

Such a set of data is plotted on Figure C-4 for a $k_b = 0.95$.

Points can now be plotted on Figure C-4 for the various turbines under consideration so as to display their relative tendencies for cavitation. The maximum discharge specific speed, N_d, is clearly dictated by the ability to limit blade surface diffusion.

N_d = DISCHARGE SPECIFIC SPEED = $\frac{NQ_3^{1/2}}{(h_{Dv})^{3/4}}$

N = ROTATING SPEED - RPM

Q_3 = ROTOR EXIT ANNULUS VOLUME FLOW - CFS

h_{Dv} = NET POSITIVE DISCHARGE HEAD - FT

= $h_{03} - h_v$ NPDH

h_{03} = ROTOR EXIT TOTAL HEAD - FT

h_v = VAPOR HEAD - FT

D_{t_2} = ROTOR TIP DIAMETER - FT

ϵ = ROTOR HUB-TIP RATIO

k_b = BLADE BLOCKAGE FACTOR - A_{FL}/A_{NN}

D_s = SUCTION SURFACE DIFFUSION FACTOR (STEWART) = $1 - \frac{V_{EXIT}}{V_{MAX}}$

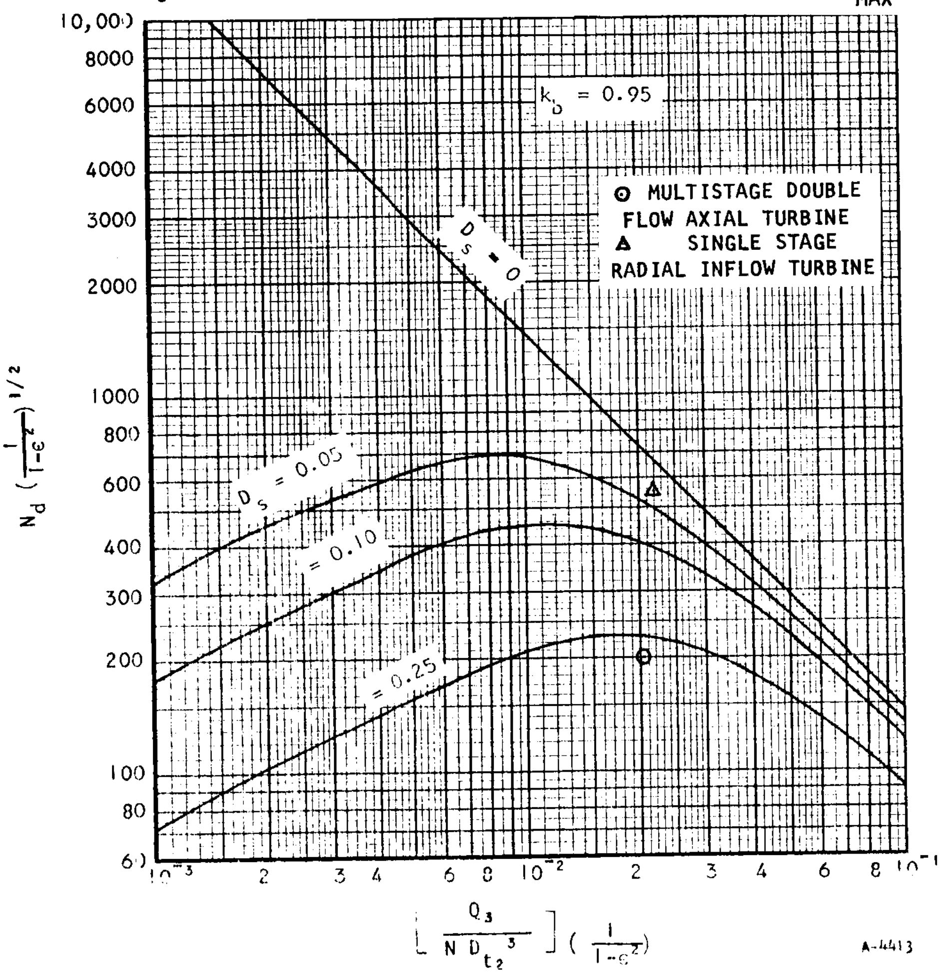


Figure C-4. Incipient Cavitation in Turbine Rotors

APPENDIX D

PRELIMINARY DYNAMIC ANALYSIS

Equations of motion representing the system dynamics may be derived by standard linearized perturbation techniques. All equations are represented in the complex domain by application of the Laplace transformation. The following sections develope the pertinent relations from which the dynamic characteristics of the system can be obtained.

a. Transient Relations--From the nature of startup control procedures, turbine acceleration rates are determined. A representative turbine map is shown in Figure D-1.

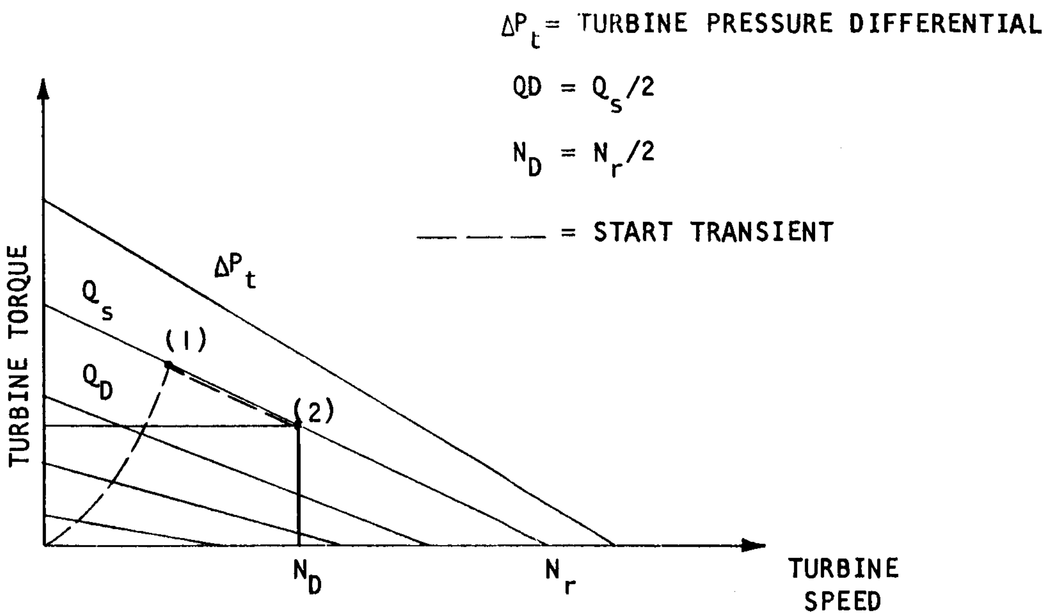


Figure D-1

Since the power adsorber is overloading during transition, an equilibrium, Point (1) is established at a reduced speed in accordance with a specific turbine pressure ratio. When the speed control is energized, the turbine moves to the design point [Point (2), Figure D-1] along a constant pressure ratio line. All control valves (except the load control valve) must operate slowly to avoid "water hammer". Turbine acceleration rates are comensurate with the slow valve operation, thus eliminating dynamic problem areas. Speed control operation at the "switch over" point will be satisfied by dynamic requirements imposed from steady-state considerations.

There are no dynamic problem areas associated with transients during failure mode operation because all valves change position slowly and the turbine merely slows down to zero speed as a function of the adsorber load.

b. Steady-State Pump Operation--A list of nomenclature used in the following section is included in Table D-1.

TABLE D-1

Lower case letters indicate perturbation quantities about the steady-state operation point.

J = Inertia, ft-lb-sec/rpm

Q = Torque, ft-lb

A = Area, ft^2

N = Speed, rpm

τ = Time constant, seconds

$\Delta P_t = P_1 - P_2$ = Turbine pressure differential

s = Subscripts apply to the set point

t = Subscripts apply to the turbine

ℓ = Subscripts apply to a loading device

c = Subscripts apply to a compressor

b = Subscripts apply to a water brake

The block diagram representing the pertinent equations of motion may be derived from the following relationships:

Basic Equation,

$$JSn = \Sigma \text{Moments} = q_t - q_\ell \tag{1}$$

Turbine,

$$Q_t = Q_s - \frac{Q_s}{N_r} N \qquad (2)$$

Where Q_s = stall torque

N_r = runaway speed (zero torque)

Load

$$Q = N^{\alpha} f(A) \qquad (3)$$

where α is a constant and f(A) is determined by the loading characteristic.

Since there are two methods of loading under consideration, (compressor and water brake) at the present time, the dynamic relations for each method may be obtained as follows.

An expression for the perturbated turbine torque (q_t) is obtained from Equation (2) and the following assumptions,

$$N_r \sim \text{Weight Flow} \sim \sqrt{\Delta P_t} \qquad (4)$$

and $$Q_s \sim \Delta P_t \qquad (5)$$

Substituting (4) and (5) into (1) which is then multiplied by N yields an expression representing the turbine power. Differentiating this expression with respect to speed and equating to zero yields peak power, a design point relationship given in Equations (6) and (7) where N_D is the design speed, and K_D a constant of proportionality.

$$N_D = K_D \sqrt{\Delta P_t} \qquad (6)$$

$$N_D = N_{r/2} \qquad (7)$$

A linearized expression for the speed set point (n_s) is obtained by differentiating Equation (6) with respect to ΔP_t and applying the "s" subscript. The result is

$$n_s = N_{D/2\Delta P_t} \qquad (8)$$

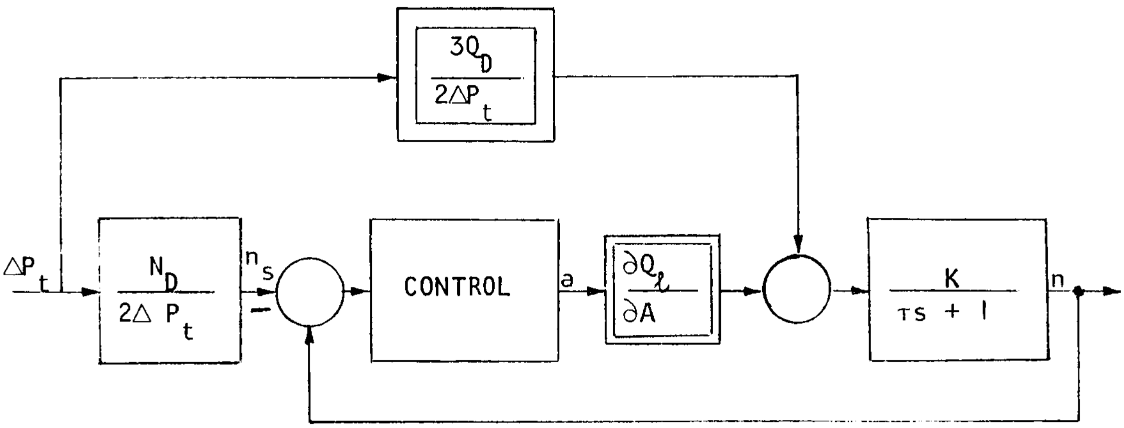
Merely substituting (4) and (5) into (2) and taking the total differential yields the desired expression for q_t, evaluated at the design point.

$$q_t = \frac{-Q_D}{N_D} n + \frac{3}{2} \frac{Q_D}{\Delta P_t} \Delta P_t \tag{9}$$

q_ℓ is obtained from the total differential of Equation (3),

$$q_\ell = \frac{\partial Q_\ell}{\partial N} n \quad \frac{\partial Q_\ell}{\partial N} a \tag{10}$$

The block diagram illustrated in Figure D-2 is a diagramatic representation of the simultaneous solution of Equations (1), (8), (9), and (10) with control provisions.



$$K = \left(\frac{Q_D}{N_D} + \frac{\partial Q_\ell}{JN}\right)^{-1} \qquad \tau = JK$$

Figure D-2

It is interesting to note that ΔP_t is a pure function of conditions upstream of the turbine and independent of speed perturbations. This condition holds for an axial flow turbine in the first approximation.

The system illustrated in Figure D-2 is evaluated at the nominal design point for controls specification. Controlled speed regulation and stability are considered for both air compressor and water brake loads. The numerical constants and load characteristics are as follows:

$$J \text{ (with air compressor)} = 9.6 \frac{\text{ft-lb-sec.}}{\text{rpm}}$$

$$J \text{ (with water brake)} = .87 \frac{\text{ft-lb-sec.}}{\text{rpm}}$$

For the Air Compressor at Design Point

$$\frac{\partial Q_c}{\partial N} = \frac{2Q_D}{N_D} \quad (\alpha = 2)$$

$$\frac{\partial Q_c}{\partial A_c} = Q_D\, \phi\, (A_c) \text{ where } \phi\, (A_c) = \frac{f'(A_c)}{f(A_c)}$$

For the Water Brake at Design Point

$$\frac{\partial Q_b}{\partial N} = \frac{2Q_D}{N_D} \quad (\alpha = 2)$$

$$\frac{\partial Q_b}{\partial A} = Q_D\, \phi\, (A_b) \text{ where } \phi\, (A_b) = \frac{f'(A_b)}{f(A_b)}$$

The system as shown in Figure D-2 has a high degree of self regulation. This property is most clearly illustrated by considering the speed response to a pressure change without control. The expression is

$$n/\Delta P_t = \frac{3/2\, Q_D\, \Delta P_t}{\frac{Q_D}{N_D} + \frac{\partial Q_\ell}{\partial N}} \tag{11}$$

The required change in speed (to maintain maximum turbine efficiency) is

$$n_s/\Delta P_t = N_D/2\Delta P_t. \tag{12}$$

If both the air compressor and water brake conform exactly to the square law assumed ($Q \sim N^2$) it is seen that

$$\frac{n}{\Delta P_t} = \frac{N_D}{2\Delta P_t} = \frac{n_s}{\Delta P_t}$$

or speed remains exactly at the required speed for peak recovery. A speed control is included in the system to provide speed regulation in the event loads do not perform exactly to the aforementioned square law.

To present a reasonable range of operational characteristics of the loads under consideration, it will be assumed that the torque to speed relationships may vary from a linear to cubic representation. The results are tabulated below:

Required:

$$n_s/\Delta P_t = \frac{N_D}{2\Delta P_t}$$

Actual:

$$n/\Delta P_t = 3/8\ N_D/\Delta P_t \text{ (cubic load)}$$

$$n/\Delta P_t = 3/4\ \frac{N_D}{\Delta P_t} \text{ (linear load)}$$

The above calculations illustrate that the control must provide the equivalent change in speed in terms of load change to make the difference between required and actual speed changes equal to zero or,

$$\frac{n_s - n}{P_t} = \left[\frac{1}{2} - \frac{3}{8}\right] \frac{N_D}{\Delta P_t} = \frac{1}{8} \frac{N_D}{\Delta P_t} \text{ (cubic load)} \qquad (13)$$

$$\frac{n_s - n}{\Delta P_t} = \left[\frac{1}{2} - \frac{3}{4}\right] \frac{N_D}{\Delta P_t} = -\frac{1}{4} \frac{N_D}{\Delta P_t} \text{ (linear load)} \qquad (14)$$

Taking the most severe condition, (14); for a given pressure change, the control must vary the load appropriately to add the equivalent of

$\frac{1}{4} \frac{N_D}{\Delta P_t}$ to the speed response obtained as a result of self regulation.

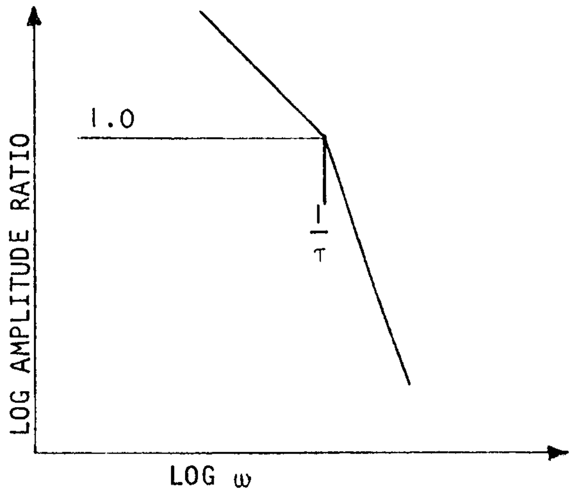
It is possible to relate speed error to turbine efficiency and hence to potential savings of the overall recovery system. Since a 1 percent turbine efficiency represents a substantial potential savings ($238,000), the functional relationship of speed error to turbine efficiency is nearly a one-to-one linear correspondence for small speed changes, and the system is ideally self regulatory, an integral mode of speed control will be introduced. The steady-state speed error will be zero and stability may be determined from the speed control loop alone.

The open loop transfer function of the speed control is

$$\frac{n}{n_s - n} = \frac{K_c K \partial Q_\ell / \partial A}{S(\tau S + 1)} \tag{15}$$

where $\dfrac{K_c\left(\dfrac{ft^2}{rpm\text{-}sec.}\right)}{S}$ is the transfer function for an integral mode controller.

The frequency response representation is as shown in Figure D-3.



$$\frac{K_c K \frac{\partial Q_\ell}{\partial A}}{T\omega^2}$$

Figure D-3

For stability, a 45° phase margin criteria will be applied. This requires that

$$\left. \frac{K_c K \dfrac{\partial Q_\ell}{\partial A}}{T\omega^2} \right|_{\omega = \frac{1}{T}} \leq 1 \tag{16}$$

Solving Equation (16)

$$K_c = \frac{1}{K\frac{\partial Q}{\partial A}} = \frac{1}{JK^2\frac{\partial Q}{\partial A}} \tag{17}$$

Equation (17) will be evaluated at the turbine design point for both methods of load absorption. These expressions represent the maximum allowable integral control gain for each of the loads. For the application, it will be desirable to reduce these gains by a factor of 2. The control gains for a square law load absorber shall be

$$K_c \text{ (air compressor)} = .47 \frac{Q_D}{N_D^2 \phi(A_c)} \frac{ft^2}{\text{rpm sec.}} \tag{18}$$

and

$$K_c \text{ (water brake)} = 5.2 \frac{Q_D}{N_D^2 \phi(A_b)} \frac{ft^2}{\text{rpm sec.}} \tag{19}$$

Since $Q_D \sim N_D^2$, the required gain is constant over the entire range except for $\phi(A_c)$ or $\phi(A_b)$ which may be taken into account by proper valve characterization.

In summary, the speed control shall be an integral type, incorporating a stability safety factor of two. The control will yield zero steady-state speed error and sufficient transient response time because all input pressure changes are slow to avoid water hammer. Taking the most severe condition, (14); for a given pressure change, the control must vary the load appropriately to add the equivalent of $\frac{1}{4}\frac{N_D}{\Delta P_t}$ to the speed response obtained as a result of self regulation.

It is possible to relate speed error to turbine efficiency and hence to potential savings of the overall recovery system. Since a 1 percent turbine efficiency represents a substantial potential savings ($238,000), the functional relationship of speed error to turbine efficiency is nearly a one-to-one linear correspondence for small speed changes, and the system is ideally self regulatory, an integral mode of speed control will be introduced. The steady-state speed error will be zero and stability may be determined from the speed control loop alone.

APPENDIX E

COST ESTIMATES

This section contains the contribution of various manufacturers on cost data related to the installation of the turbine and heat exchanger recovery systems. The problem statements were based upon preliminary design type concepts and were modified as deemed appropriate by the contributors. The results are therefore budgetary type estimates. In some cases, the estimates are "raw" or "bare costs"; in others contingency and profit factors have been added.

Heat Exchanger Cost Estimate (AiResearch Manufacturing Company)

A budgetary estimate is supplied below. This estimate is based on a finned tube configuration using existing commercially available tubes in a four-pass cross-counterflow arrangement. The tube bundle can be folded to give two banks of 5300 tubes 18 feet long.

The cost of building this unit at AiResearch would necessarily include non-recurring charges associated with handling and processing components of unprecedented size and weight. The cost of AiResearch to obtain the required equipment such as hoists, in plant trucks, cleaning baths, etc., is estimated to be $100,000. This does not include large machine and forming tools. Machining and forming would be subcontracted.

The cost of designing and fabricating one unit is broken down as follows:

Engineering	$ 80,000
Laboratory	10,000
Facilities	10,000
Shop	100,000
Tooling	50,000
Purchased Parts (Machining, etc.)	78,000
Material (including finned tube)	96,000
	$ 424,000

These costs are predicated on a very limited test program to verify performance estimates as deemed necessary. The basic premise is that the unit will be sufficiently conservative in design to assure that the performance requirements will be met. Testing costs can be kept at a minimum by using this approach, although the most compact unit will not be obtained.

The above estimate does not include:

(1) Insulation
(2) General and Administrative Expense
(3) Fee

If a contract is sought to build this unit, it should be split into a design phase and a fabrication phase, with each phase negotiated separately. An adequate cost estimate on fabrication can more reasonably be made after a design is available.

With G and A and fee included, the estimated price for this heat exchanger is:

Fabrication	$ 424,000
Non-recurring Charges	100,000
	$ 524,000
G & A (13.6%)	71,264
	$ 595,264
Fee (7%)	$ 41,668
Insulation	$ 20,000
	$ 656,932

4 header plates	weight
3.5 ft x 8 ft x 3.0 in.	
5300 holes .375 inch dia. on 1.0	
inch spacing 4 x 3 1/2 x 8 x 1/6 = 175 =	3,300 lbs.

200, 000 feet of finned tube 0. 154 lbs/ft

wolverine tru fin hr 63-0906035-41 (@ \$. 25 per foot = \$50, 000)	30, 800 lbs

4 high pressure pans

4 x 3 1/2 x 8 x 1/4 = 28 cu ft x 175 = 3. 5 ft x 8 ft x 3. 0 in. with 2. 0 ft radius	4, 900 lbs

2 low pressure pans

8. 0 ft x 18. 0 ft x 0. 5 inch formed to 6. 0 ft radius 2 x 6 x 1/24 = 12 cu ft x 175 =	2, 100 lbs

inlet, outlet and crossover tubes

50 ft x 10 inch pipe 3 x 50 = 150 sq ft x 1/12 = 12 1/2 cu ft x 175 =	2, 200 lbs

baffles - 10 2' spacing

made in strips 100 pieces 0. 75 in x . 375 in x 8. 0 ft 100 28/12 x 8 = 18 cu ft x 175 =	3, 300 lbs
miscellaneous brackets and structure	4, 000 lbs
total weight	50, 000 lbs
vacuum jacket-tank	15, 000 lbs

(\$20, 000 to \$40, 000)

Installation Cost Estimates

The following installation cost estimates were based upon the schematic diagrams shown at various places in the rest of this report. Those estimates include control valves, and in two instances, the specific proposal by Hammel-Dahl Company, which is included in this section.

LINDE COMPANY

DIVISION OF UNION CARBIDE CORPORATION

2770 LEONIS BOULEVARD, LOS ANGELES 58, CALIFORNIA

December 12, 1963

Airesearch Manufacturing Company
9851 Sepulveda Blvd.
Los Angeles 9, California

Attention: Mr. H. G. Starck
Preliminary Design, Dept. 933

Gentlemen:

This has reference to our discussion regarding the design of a liquid to vapor hydrogen heat exchanger that concerns your study for Aerojet-General on the M-1 Program.

We have reviewed your requirements and estimate that the cost of the heat exchanger will be between $200,000 and $300,000. We have not attempted to make an engineering analysis of the heat exchanger design per your performance requirements in an attempt to reduce cost or equipment size. Should this heat exchanger become a reality, the Linde Division would be pleased to perform an engineering design and quote on the equipment. The cost noted above, however, should be fairly realistic at the present time. If we can provide any additional information, please advise.

Very truly yours,

UNION CARBIDE CORPORATION
Linde Division

P. J. Murto
Cryogenic Products Department
Pacific Coast Region

PJM/ect

FORM 114 COL. P.

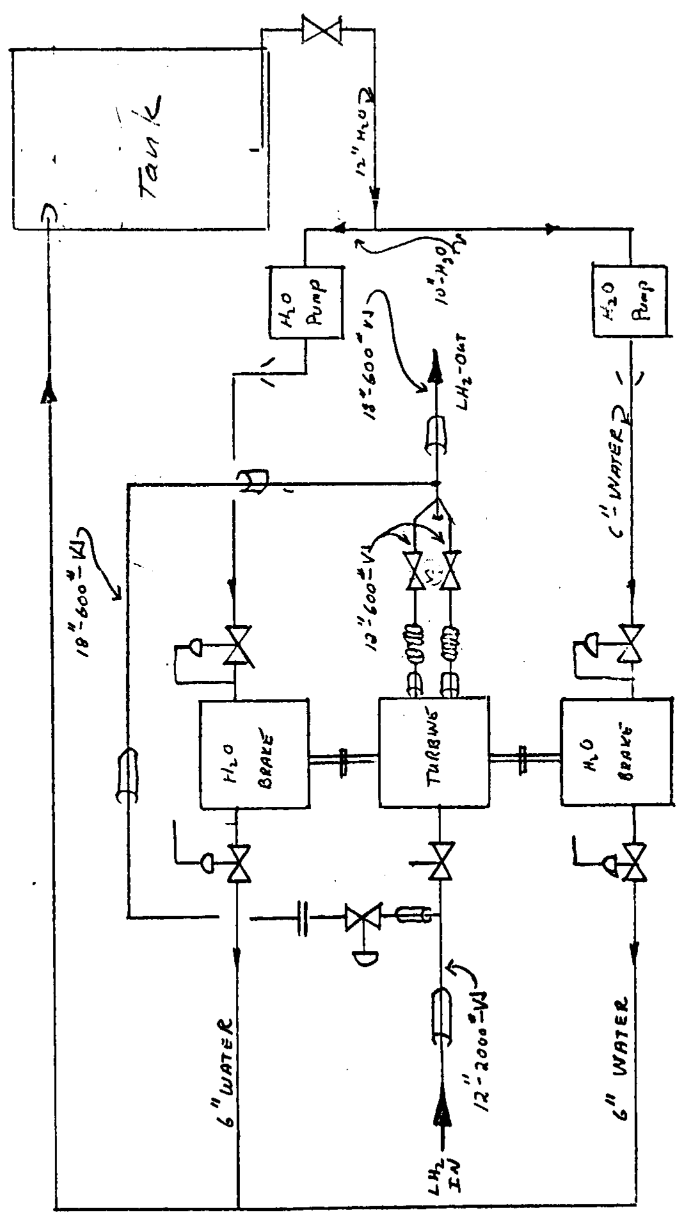
Stearns-Roger CORPORATION

PAGE

JOB NO. DATE 1/9/64 BY CH'K

CUSTOMER PROJECT

SUBJECT



Schematic Diagram of Turbine - Water ...

FORM 11-C 3-63

Stearns-Roger
CORPORATION

ESTIMATE NO. Budget — SHEET 1 OF ____

CUSTOMER Garrett - AiResearch Manufacturing Company — DATE 1/9/64

JOB LH_2 Recovery Project - HT-Exchanger Method — BY WRL/WNB

	WEIGHT OR QUAN.	HOURS	LABOR	MATERIAL	OTHER	TOTAL
Excavation & Sitework						$ 3,000.00
Concrete	10 yds					2,000.00
Structural						30,000.00
Piping						140,000.00
Radiograph						1,000.00
Checkout & Test						8,000.00
Controls						8,100.00
Heat Exchanger (Installation)	30 tons					4,000.00
Freight & Handling						7,000.00
Sub Total Raw Costs						203,100.00
Engineering						25,000.00
Field Expense (Contractor)						20,000.00
						248,100.00
Contingency						25,000.00
Profit						15,000.00
Total Price Less Heat Exchanger						$288,100.00

FORM F. 3-63

Stearns-Roger
CORPORATION

ESTIMATE NO. Budget | SHEET ____ OF ____
CUSTOMER Garrett - AiResearch Manufacturing Company | DATE 1/9/64
JOB Turbo Expander System | BY WRL/WNB

	WEIGHT OR QUAN.	HOURS	LABOR	MATERIAL	OTHER	TOTAL
Hydrogen Loop						
Sitework						$ 1,200.00
Concrete						3,000.00
Piping						76,900.00
Testing & X-rays						4,500.00
Instrumentation						8,100.00
Freight & Handling						9,000.00
Total Hydrogen Loop						$102,700.00
Water Loop						
Sitework						900.00
Concrete						300.00
Piping						14,297.00
Tank						5,000.00
Elec. & Inst.						8,000.00
Water Pumps (2)						6,200.00
Installing Brake						2,500.00
Installing Expander						5,000.00
Total Water System						$42,197.00
Subtotal Raw Cost						$144,897.00

1084 F. 3-63

Stearns-Roger
CORPORATION

ESTIMATE NO. Budget — SHEET 2 OF 2

CUSTOMER Garrett - AiResearch Manufacturing Company — DATE 1/9/64

JOB Turbo Expander System — BY WRL/WNB

	WEIGHT OR QUAN.	HOURS	LABOR	MATERIAL	OTHER	TOTAL
Engineering						$ 25,000.00
Field Expense (Contractor)						15,000.00
						184,897.00
Contingency						18,900.00
Profit						20,000.00
Total Price Less Brakes & Expanders						223,797.00

C F BRAUN & CO

ESTIMATE

CUSTOMER Airesearch — PROJECT 3700

PAGES 1 — PAGE 1

UNIT Hydrogen Recovery - Hydraulic Loader — DATE 2-20-64 — BY RLR

ITEM	QUANTITY	UNIT COST MATL	UNIT COST LABOR	TOTAL COST MATERIAL	TOTAL COST FIELD-LABOR
SUMMARY					
Water Tanks	2			$ 2,000	1,000
Install Water Pumps	2			-	2,000
Install Hydraulic Loader	1			-	2,000
Instrumentation	SAY			4,000	1,000
Civil Work	100 CY			5,000	10,000
Piping *				185,000	23,000
Electrical				6,000	10,000
				$ 202,000	49,000
				49,000	
				$ 251,000	
Construction Indirects				60,000	
				311,000	
				300,000	

Note - Price of Expander, Water Brakes, Pumps and Motors not included

* Including Hammel -Dahl control valves

C F BRAUN & CO

ESTIMATE

CUSTOMER Airesearch — PROJECT 3700

PAGES 1 — PAGE 1

UNIT Hydrogen Recovery - Exchanger — DATE 2-20-64 — BY RLR

ITEM	QUANTITY	UNIT COST MATL	UNIT COST LABOR	TOTAL COST MATERIAL	TOTAL COST FIELD-LABOR
SUMMARY					
Civil Work				1,000	5,000
Structural Steel				30,000	5,000
Piping				175,000	2,5000
Insulation (40 feet of 16 inch 3 inch thick Polyruethane)				6,000	-
Install heat exchanger				-	1,000
				212,000	36,000
				36,000	
				$ 248,000	
Construction Indirects				62,000	
				$ 310,000	
				300,000	
Note - Price of Heat Exchanger not included					

PAUL HARDEMAN, INC.

OPERATION: ESTIMATE SUMMARY — HEAT EXCHANGER SYSTEM
ITEM NO. — SHEET 1 OF 1 PAGE 1
JOB: AIR RESEARCH
EST. BY: LA FORTE — CK'D BY — DATE: 2/26/64

	QUANTITY	UNIT	LABOR AND INSURANCE UNIT COST	LABOR AND INSURANCE AMOUNT	PERMANENT MATERIAL UNIT COST	PERMANENT MATERIAL AMOUNT	SPECIFIC PLANT UNIT COST	SPECIFIC PLANT AMOUNT	EQUIPMENT RENTAL UNIT COST	EQUIPMENT RENTAL AMOUNT	SUPPLIES UNIT COST	SUPPLIES AMOUNT	SUB-CONTRACTS UNIT COST	SUB-CONTRACTS AMOUNT	TOTAL DIRECT COST UNIT COST	TOTAL DIRECT COST AMOUNT
FURNISH & INSTALL SUPPORTS FOR & INSTALL HEAT EXCHANGER				—		20000		—		—		—		15000		35000
REMOVAL OF EXISTING PIPING, VALVES, ETC.	184	M/HR	6^{50}	1196		—		—	10% L	120	5% L	60		—		1376
VJ PIPING, FAB., MAT'LS				—		40503		—		—		—		—		40503
LABOR				—		3656		—		—		—		—		8655
MISC.				—		2464		—		—		—		—		2464
REGULAR PIPING, FAB, MATLS				—		10437		—		—		—		—		10427
LABOR				—		1834		—		—		—		—		1834
MISC.				—		1141		—		—		—		—		1141
FIELD INSTALLATION MAT'LS	442	M/HR	6^{50}	2873		49611		—	10% L	287	5% L	144		—		52915
INSTALL SPOOLS & FIELD WELDS	271	M/HR	6^{50}	1762		—		—	10% L	176	5% L	88		—		2026
FIELD INSULATE VJ PIPING JOINTS	60	M/HR	6^{50}	390		2200		—	10% L	39	5% L	20		2200		4849
FIELD MISC. (TEST, CLEAN PAINT)	64	M/HR	6^{50}	416		—		—		150		375		500		1441
SUB TOTALS				6637		136535		—		772		687		17700		162631
(SUPERVISION, COMMUNICATIONS, TRIPS, ETC.) GENERAL CONDITIONS	ALLOW			2000		—		—		500		1000		—		3500
SUB TOTALS				8637		136835		—		1272		1687		17700		166131
SALES TAXES (SAY 4%)				—		5473		—		—		67		—		5540
SUB TOTALS				8637		142308		—		1272		1754		17700		171671
												OVERHEAD 5%				8584
												SUB TOTAL				180255
												FEE 5%				9013
												TOTAL BID				189268

PAUL HARDEMAN, INC. — MECH. ESTIMATING DETAIL SHEET. — PAGE 1

JOB AIR RESEARCH — SHT 1 OF 1

DESCRIPTION HEAT EXCHANGER SYSTEM

Description	Quantity	Material Unit Price	Material Extension	Extension	Labor Unit	Labor Extension
FIELD INSTALLATION MAT'LS RE-CAP						
FROM PG. NO. SHT. 1 OF 2				49275.00		364 —
" " " " 2 OF 2				336.00		78 —

JOB AIR RESEARCH — SHT 2 OF 2

DESCRIPTION HEAT EXCHANGER SYSTEM

Description	Quantity	Material Unit Price	Material Extension	Extension	Labor Unit	Labor Extension
FIELD INSTALLATION MAT'LS (CONT'D)						
SUPPORTS FOR PIPING, NEW & ALTERATIONS	1 LOT	ALLOW		300.00		60 —
B N & GASK SETS 150# C.S. 16"	3 SETS	ALLOW 12.00		36.00	6 —	18 —

DATE: 2/25/64 SPEC. SECTION TOTAL 49,611.00 TOTAL 442

EST. BY LA FORTE PRICED BY: CHECKED BY:

PH-377

PAUL HARDEMAN, INC.

MECH. ESTIMATING DETAIL SHEET.

JOB AIR RESEARCH — SHT 1 OF 2

DESCRIPTION HEAT EXCHANGER SYSTEM

		MATERIAL				LABOR	
Item	Size	QUANTITY	UNIT PRICE	EXTENSION	EXTENSION	UNIT	EXTEN.
FIELD INSTALLATION MAT'LS •							
THROTTLE VALVE, 300# FL'GD, S/S, LH_2 SERVICE,	12"	1 EA.	ALLOW 5000.00		5000.00	8-	8-
SHUT-OFF VALVE, 300# FL'GD, S/S, LH_2 SERVICE,	18"	1 "	REUSE EXIST. VALVE			16-	16-
	18"	3 "	ALLOW 8000.00		24000.00	16-	48-
SHUT-OFF VALVE, 150# FL'GD, S/S GH_2 SERVICE,	18"	1 "	ALLOW 6000.00	6000.00		9-	9-
	24"	1 "	10000.00	10000.00	16000.00	16-	16-
RESTRICTION ORIFICE, S/S, (PLATES ONLY)	18"	1 "	REUSE EXIST. ORIFICE			2-	2-
EXPANSION JTS, S/S, 300# FL'GD,	18"	3 "	ALLOW 1300.00		3900.00	16-	48-
STUD, NUT & GASK. SETS, (S/S STUDS & NUTS) (TEFLON RING GASK.) 300#	12"	2 SETS	ALLOW 9.00	18.00		6-	12-
"	18"	18 "	14.00	252.00		9-	162-
150#	18"	3 "	14.00	42.00		6-	18-
"	24"	3 "	20.00	60.00	372.00	8-	24-
EXTRA TEFLON GASK., 300#	18"	1 EA.	ALLOW 3.00		3.00	1-	1-
				TOTAL	49,275.00	TOTAL	364 HR

DATE: 2/28/64 SPEC. SECTION

EST. BY LA FORTE PRICED BY: CHECKED BY:

PH-277

PAUL HARDEMAN, INC.

OPERATION ESTIMATE SUMMARY — ITEM NO. — SHEET 1 OF 1 PAGE 20

HYDROGEN LOOP — JOB AIR RESEARCH

QUANTITY — UNIT — EST. BY LAFORTE — CK'D BY — DATE 2/21/64

	QUANTITY	UNIT	LABOR AND INSURANCE UNIT COST	LABOR AND INSURANCE AMOUNT	PERMANENT MATERIALS UNIT COST	PERMANENT MATERIALS AMOUNT	SPECIFIC PLANT UNIT COST	SPECIFIC PLANT AMOUNT	EQUIPMENT RENTAL UNIT COST	EQUIPMENT RENTAL AMOUNT	SUPPLIES UNIT COST	SUPPLIES AMOUNT	SUB-CONTRACTS UNIT COST	SUB-CONTRACTS AMOUNT	TOTAL DIRECT COST UNIT COST	TOTAL DIRECT COST AMOUNT
FURNISH CONC. PAD FOR TURBINE & POWER ABSORBERS; INSTALL TURBINE	72	M/HR	6.50	468		100		—	10% L	47	5% L	23		300		938
REMOVE EXISTING PIPING	32	M/HR	6.50	208		—		—	10% L	21	5% L	10		—		239
VJ PIPING, FAB., MAT'LS				—		45632		—		—		—		—		45632
LABOR				—		10659		—		—		—		—		10659
MISC.				—		2464		—		—		—		—		2464
FIELD INSTALLATION MAT'LS	272	M/HR	6.50	1768		36360		—	10% L	177	5% L	88		—		38393
INSTALL SPOOLS & FIELD WELDS	245	M/HR	6.50	1593		—		—	10% L	159	5% L	80		—		1832
FIELD INSULATE VJ PIPING JTS.	60	M/HR	6.50	390		2200		—	10% L	39	5% L	20		1100		3749
FIELD MISC. (TEST, CLEAN)	64	M/HR	6.50	416		—		—		150		375		300		1241
SUB TOTALS				4843		97415		—		593		596		1700		105647
GENERAL CONDITIONS	ALLOW			2000		—		—		500		1000		—		3500
SUB TOTALS				6843		97415		—		1043		1596		1700		109147
SALES TAXES (SAY 4%)				—		3917		—		—		64		—		3981
SUB TOTALS				6843		101332		—		1093		1660		1700		113128
												OVERHEAD		5%		5656
												SUB TOTAL				118784
												FEE 5%				5939
												TOTAL BID				124723

PAUL HARDEMAN, INC. — MECH. ESTIMATING DETAIL SHEET.

JOB: AIR RESEARCH — SHT 1 OF 2

DESCRIPTION: HYDROGEN LOOP

FIELD INSTALLATION MAT'LS

Description	Size	Quantity	Material Unit Price	Material Extension	Material Extension	Labor Unit	Labor Extension
THROTTLE VALVE, 300# FL'G'D, S/S, LH_2 SERVICE							
	12"	2 EA.	ALLOW 5000.00		10000.00	8-	16-
SHUT-OFF VALVES, S/S, LH_2 SERVICE,							
300# FL'G'D,	12"	1 "	ALLOW 5000.00	5000.00		8-	8-
WRONG (REVERSE) 900# FL'G'D,	12"	2 "	ALLOW 9000.00	18000.00		10-	20-
				→	23000.00		
RESTRICTION ORIFICE, S/S, (PLATES ONLY)	18"	1 "	REUSE	EXIST.	ORIFICE	2-	2-
	18"	1 "	ALLOW		300.00	2-	2-
EXPANSION JTS, S/S, 300# FL'G'D	12"	2 "	ALLOW 900.00	1800.00		8-	16-
900# FL'G'D	12"	1 "	ALLOW 1200.00	1200.00		10-	10-
				→	3000.00		
STUD, NUT & GASK SETS (S/S STUDS & NUTS)							
(TEFLON RING GASK.), 300#,	12"	8 SETS	ALLOW 9.00	72.00		6-	48-
	18"	5 "	ALLOW 14.00	70.00		9-	45-
				→	142.00		
STUD, NUT & GASK SETS (S/S STUDS & NUTS)							
(S/S OCT. R.T.J. GASK.), 900#	12"	6 "	ALLOW 25.00	150.00		8-	48-
	18"	1 "	ALLOW 40.00	40.00		12-	12-
				→	190.00		
				TOTAL	36632.00	TOTAL	227 HR

DATE: 2/28/64 SPEC. SECTION

EST. BY LA FORTE PRICED BY: CHECKED BY:

PH-777

PAUL HARDEMAN, INC. MECH. ESTIMATING DETAIL SHEET.

JOB AIR RESEARCH SHT 2 OF 2

DESCRIPTION HYDROGEN LOOP

	MATERIAL				LABOR	
	QUANTITY	UNIT PRICE	EXTENSION	EXTENSION	UNIT	EXTENSION
FIELD INSTALLATION MAT'LS (CONT'D)						
EXTRA GASK., TEFLON RING TYPE, 300#, 18"	1 EA.	Allow 3.00	3.00		2-	2-
S/S OCT. RTJ 900#, 18"	1 "	Allow 25.00	25.00		3-	3-
				28.00		
SUPPORTS	1 LOT	ALLOW		200.00		40-

JOB AIR RESEARCH SHT 1 OF 1

DESCRIPTION HYDROGEN LOOP

	MATERIAL				LABOR	
	QUANTITY	UNIT PRICE	EXTENSION	EXTENSION	UNIT	EXTENSION
FIELD INSTALLATION MAT'LS RE-CAP						
FROM PG. NO. SHT. 1 OF 2				36632.00		227-
" " " " 2 OF 2				228.00		45-
TOTAL				36860.00	TOTAL	272 HR

DATE: 2/28/64 SPEC. SECTION

EST. BY LA FORTE PRICED BY: CHECKED BY:

PH-277

PAUL HARDEMAN, INC.

OPERATION ESTIMATE SUMMARY — ITEM NO. — SHEET 1 OF 1 PAGE 34

WATER LOOP — JOB AIR RESEARCH

QUANTITY — UNIT — EST. BY LA FORTE — CK'D BY — DATE 2/28/64

	QUANTITY	UNIT	LABOR AND INSURANCE UNIT COST	LABOR AND INSURANCE AMOUNT	PERMANENT MATERIALS UNIT COST	PERMANENT MATERIALS AMOUNT	SPECIFIC PLANT UNIT COST	SPECIFIC PLANT AMOUNT	EQUIPMENT RENTAL UNIT COST	EQUIPMENT RENTAL AMOUNT	SUPPLIES UNIT COST	SUPPLIES AMOUNT	SUB-CONTRACTS UNIT COST	SUB-CONTRACTS AMOUNT	TOTAL DIRECT COST UNIT COST	TOTAL DIRECT COST AMOUNT
FURNISH & INSTALL WATER TANK				—		20		—		—		—		2200		4200
FURNISH & INSTALL BRAKES & WATER PUMPS (CONC. PADS FOR BRAKES IS INCLUDED WITH LH_2 LOOP)	264	M/HR	6.50	1716		140		—	10% L	172	5% L	86		200		21174
REGULAR PIPING, FAB, MAT'LS				—		140		—		—		—		—		1401
LABOR				—		94		—		—		—		—		941
MISC				—		99		—		—		—		—		995
FIELD INSTALLATION MAT'LS	163	M/HR	6.50	1060		493		—	10% L	106	5% L	53		—		6155
INSTALL SPOOLS & FIELD WELDS	47	M/HR	6.50	306		—		—	10% L	31	5% L	15		—		352
FIELD MISC. (TEST, CLEAN, PAINT)	32	M/HR	6.50	208		—		—		200		125		300		833
SUB TOTALS				3290		24273		—		509		279		2700		36051
GENERAL CONDITIONS	ALLOW			2000		—		—		500		1000		—		3500
SUB TOTALS				5290		29273		—		1009		1279		2700		39551
SALES TAXES (SAY 4%)				—		1171		—		—		51		—		1222
SUB TOTALS				5290		30444		—		1009		1330		2700		40773
														OVERHEAD 5%		2039
														SUB TOTAL		42812
														FEE 5%		2141
														TOTAL BID		44953

PAUL HARDEMAN, INC. — MECH. ESTIMATING DETAIL SHEET. — PAGE 40

JOB AIR RESEARCH — SHT 1 OF 1

DESCRIPTION WATER LOOP

		MATERIAL				LABOR	
		QUANTITY	UNIT PRICE	EXTENSION	EXTENSION	UNIT	EXTENSION
FIELD INSTALLATION MAT'LS							
PRESSURE CONTROL VALVES, 150# FL'GD, c/s,	6"	2 EA.	ALLOW 800^{00}		1600^{00}	8-	16-
GATE VALVES, 150# FL'GD, c/s	6"	4 "	410^{00}	840^{00}		3-	12-
	10"	2 "	932^{00}	1864^{00}		4-	8-
	12"	1 "	1210^{00}	1210^{00}		5-	5-
			- 40	3914^{00}	2348^{40}		
CHECK VALVES, 150# FL'GD, c/s,	6"	2 "	400^{00}	-40	480^{00}	3-	6-
PRESS GAGES, SWITCHES TB'G, BLOCK VALVE ETC.		1 LOT ALLOW			300^{00}		48-
BOLT NUT & GASK SETS, 150#	6"	11 SETS	ALLOW 4^{00}	44^{00}		2-	22-
	8"	2 "	6^{00}	12^{00}		3-	6-
	10"	4 "	8^{00}	32^{00}		4-	16-
	12"	2 "	10^{00}	20^{00}		4-	8-
					108^{00}		
SUPPORTS		1 LOT	ALLOW		100^{00}		16-
				TOTAL	4936^{40}	TOTAL	163 HR

DATE: 2/28/64 SPEC. SECTION

EST. BY LA FORTE PRICED BY: CHECKED BY:

PH-277

PAUL HARDEMAN, INC. MECH. ESTIMATING DETAIL SHEET. PAGE 36

JOB Air Research SHT 1 OF 1

DESCRIPTION Water Loop

Furnish & Install Brakes & Water Pumps (Conc. pad for brake is included with LH_2 Loop)

Description	Material: Quantity	Material: Unit Price	Material: Extension	Material: Extension	Labor: Unit	Labor: Extension
30,000 HP Water Brakes, Kahn	2 EA	Allow 4500.00		9000.00		
061-040-001			Labor			48-
DBL Suction Water Pumps, 3000 GPM,	2 "	Allow 4500.00		9000.00		
150 HP, 1750 RPM, De Laval			Labor			96-
Conc. Pads for Pumps	2 "	Allow	SUB 200.00	—	—	—
Electrical Work	1 Lot	Allow		1000.00		
			Labor			120-
	1	SUB 200.00	1 TOTAL	19,000.00	TOTAL	264 HR

DATE: 2/28/64 SPEC. SECTION

EST. BY La Forte PRICED BY: CHECKED BY:

94-277

AIRESEARCH - LIQUID HYDROGEN RECOVERY PROJECT

ESTIMATED INSTALLED COST - CASE I
Turbo Hydraulic Expander Installation

		Material	Labor	Total
I	Site Work	1,000	600	1,600
II	Concrete and Foundations	4,500	6,000	10,500
III	Radiographic and Mass Spec.	6,000		6,000
IV	Insulation	18,000		18,000
V	Purge and Test	6,000	5,000	11,000
VI	Mechanical Check-out		4,000	4,000
VII	Piping - low temperature 304 SS			
	Fittings	25,800	14,800	40,600
	Valves	208,000	7,000	215,000
	Pipe	10,400	4,000	14,400
	Screwed Fittings	1,200	2,300	3,500
	Vacuum Jacketing	7,200	5,700	12,900
VIII	Water Piping			
	Fittings	2,000	2,000	4,000
	Pipe	2,000	1,000	3,000
	Strainers	2,500	200	2,700
	Valves	1,700	400	2,100
IX	Hydra;ulic Piping	2,400	600	3,000
X	Controls	13,500	1,400	14,900
XI	Tanks	9,000		9,000
XII	Equipment			
	Turbo Expander-Hydraulic (AiResearch)		17,000	17,000
	2 Kahn water brakes, 30,000 HP ea.	150,000		150,000
	Centrifugal water pump, I-R AVS	5,000	1,200	6,200
XIII	Freight and handling	12,000		12,000
		488,200	73,200	561,400

Turbo Hydraulic Expander Installation

		Material	Labor	Total
XIV	Contractor's Indirect Expense			
	Materials, Supplies, misc.	36,000		
	Supervision		12,000	
	Labor Burden		12,000	
		526,200	97,200	623,400
	Contingency - 5%	25,000	5,000	30,000
		551,200	102,200	653,400
	Contractor's Profit - 10%			64,000
				717,400
	Engineering and Purchasing at 5%			34,000
				$752,400

ESTIMATED INSTALLED COST - CASE II
Heat Exchange Alternative

AIRESEARCH - LIQUID HYDROGEN RECOVERY PROJECT

		Material	Labor	Total
I	Site work	2,000	1,000	3,000
II	Concrete (60 cu. yds.)	2,400	3,600	6,000
III	Radiographic Inspection (sub)	6,000	4,000	10,000
IV	Structural	34,000	16,000	50,000
V	Purge and Test	4,000	4,000	8,000
VI	Mechanical Check-out		4,000	4,000
VII	Piping - low temperature 304 SS			
	Fittings	40,600	13,000	
	Pipe	45,300	3,600	
	Valves	74,000	3,400	
	Expansion Joints	24,000	1,800	
	Vacuum Jacketing	11,700	5,200	
	Weld and fabrication	600	16,000	
	Misc. pipe supports	2,500	1,200	
		199,700	39,200	219,900
VIII	Controls	6,500	1,600	8,100
IX	Equipment			
	Heat Exchanger (AiResearch)		6,000	6,000
X	Freight and Handling	14,000		
XI	Insulation	18,000		
		286,600	78,400	365,000
XII	Contractor's Indirect Expense			
	Materials, supplies and equipment rental, etc.	35,000		35,000
	Supervision		12,000	12,000
	Labor Burden		12,000	12,000
		321,600	102,400	424,000
	Contingency - 10%	30,000	10,000	40,000
	Engineering and Purchasing at 5%			25,000
	Contractor's Profit - 10%			50,000
				539,000

ITT GENERAL CONTROLS INC.
A SUBSIDIARY OF INTERNATIONAL TELEPHONE AND TELEGRAPH CORPORATION

HAMMEL-DAHL/FOSTER DIVISION
801 ALLEN AVENUE, GLENDALE, CAL. 91201—TELEPHONE (213) 849-2348

PROPOSAL

TO

AIRESEARCH MANUFACTURING COMPANY

BY

ITT GENERAL CONTROLS INC.
HAMMEL-DAHL/FOSTER DIVISION

SUMMARY

HAMMEL-DAHL CONTROL VALVES

Liquid Hydrogen Loop Valves		
(1) 12" - 900 LB ASA	35,000	
(1) 12" - 900 LB ASA	33,000	
(2) 12" - 150 LB ASA	36,000	
(1) 12" - 300 LB ASA	20,000	
	$124,000	
(2) Controllers	11,000	
		135,000
Check valves	120	
Filters	1,800	
Accumulators	4,590	
	6,510	
Total LH_2 Loop		
		$141,510
Water Loop Valves		
(4) 6" 150 LB ASA		$4,400
Heat Exchange System		
(1) 12" - 300 LB ASA		20,000
(1) 18" - 150 LB ASA		40,000
(1) 1 Controller		5,500
		$65,500

LIST OF CONTROL VALVES

1. One (1) -12" Hammel-Dahl Stainless Steel Control Valve. Hydraulically operated angle valve, butt weld connections, 900# ASA rated, stainless steel body and trim, linear characteristic, CV=1400, design CV=1260 plus/minus 10%, balanced construction, flash-flo design, extension bonnet with teflon packing. Valve to be complete with hydraulic cylinder, mechanical latch, servo valve manifold and potentiometer and limit switches.

   ESTIMATED PRICE EACH $35,000.00

2. One (1) -12" Hammel-Dahl Stainless Steel Shut-Off Valve. Hydraulically operated angle valve, butt weld connections, 900# ASA rated, stainless steel body and trim, linear characteristic, CV=1400, design CV=1260 plus/minus 10%, balanced construction, flash-flo design, extension bonnet with teflon packing. Valve to be complete with hydraulic cylinder, mechanical latch, servo valve manifold and potentiometer and limit switches.

   ESTIMATED PRICE EACH $33,000.00

3. One (1) - 12" Hammel-Dahl Stainless Steel Control Valve. Hydraulically operated angle valve, butt weld connections, 300# ASA rated, stainless steel body and trim, linear characteristic, CV=1400, design CV=1260 plus/minus 10%, balanced construction, flash-flo design, extension bonnet with teflon packing. Valve to be complete with hydraulic cylinder, mechanical latch, servo valve manifold and potentiometer and limit switches.

   ESTIMATED PRICE EACH $20,000.00

4. Two (2) - 12" Hammel-Dahl Stainless Steel Shut-Off Valves. Hydraulically operated angle valve, butt weld connections, 150# ASA rated, stainless steel body and trim, linear characteristic, CV=1400, design CV=1260 plus/minus 10%, balanced construction, flash-flo design, extension bonnet with teflon packing. Valve to be complete with hydraulic cylinder, mechanical latch, servo valve manifold and potentiometer and limit switches.

ESTIMATED PRICE EACH $18,000.00

5. One (1) - 18" Hammel-Dahl Stainless Steel Shut-Off Valve. Hydraulically operated angle valve, butt weld connections, 150# ASA rated, stainless steel body and trim, linear characteristic, CV=1400, design CV=1260 plus/minus 10%, balanced construction, flash-flo design, extension bonnet with teflon packing. Valve to be complete with hydraulic cylinder, mechanical latch, servo valve manifold and potentiometer and limit switches.

ESTIMATED PRICE EACH $40,000.00

6. Four (4) - 6" Hammel-Dahl Model A20V800 Cast Steel Control Valves, 150# ASA rated.

ESTIMATED PRICE EACH $ 1,100.50

Valve performance shall be as follows:

Valves shall stroke from full-closed to full-open in not more than 1.0 second, and from full-open to full-closed in not more than 1.0 second, including snubbing time. Hydraulic snubbing may be used within the last 5 per cent of the actuator travel limits. Frequency response shall be at least 1 cycle per second at a phase shift of 45 degrees with an input signal range of plus or minus 10 per cent.

ACCESSORY EQUIPMENT

1. Accumulators - Three (3) accumulators will be furnished, one for each two valves. Each accumulator shall be sized to stroke the control valves at least six times after failure of electrical power with a minimum supply pressure of 2000 psig.

   Three (3) - Stainless Steel Accumulators, 9-gallon capacity, for use with Skydrol 500A.

   NET PRICE EACH $1,530.00

2. Filters - Six (6) filters will be furnished, one for each control valve. Each filter will be sized so that it can withstand full Delta P without any damage to the filter element. The filter element will be rated a 5 micron capability.

   Six (6) - Filters, 10 gpm, 5 micron, nominal rated.

   NET PRICE EACH $ 300.00

3. Check Valves - Three (3) check valves will be furnished, one for each accumulator.

   Three (3) - Check valves.

   NET PRICE EACH $ 40.00

ELECTRONIC CONTROLLERS

The Controller that we recommend is the Micro Gee Model 57S Controller modified as necessary to meet the specific job requirements which are not completely defined at this time.

The Micro Gee 57S Controller is one of the 57 series Pressure Controllers. This Controller is designed to work with strain bridge type transducers, (which it is assumed at this point will be used for this application).

The 57S Controller operates in the following manner.

A pressure transducer senses the actual pressure downstream of the pressure reducing valve and transmits a proportional signal to the pressure controller for comparison with the signal of the set point control. The controller modifies the difference between these signals in accordance with the static and dynamic requirements of the system and transmits the signal to the servo assembly which is coupled to the pressure reducing valve servo operator. The servo operator positions the pressure reducing valve in proportion to these signals.

When properly adjusted, the actual pressure and the set point pressure will agree within the tolerances of the pressure transducer and indicators used. The pressure controller consists of the following functional components.

(1) A Setpoint Control, which is part of a summing circuit to be compared with the output of the pre-amplifier.

(2) A Pre-Amplifier to raise the voltage level of any difference (error) signal. The gain of this stage is fixed.

(3) Two D. C. Amplifiers around which gain, rate and integration networks may be added to modify the error signal to minimize transient oscillations and response time and reduce the steady state error to zero.

(4) Capabilities of remote set point and ext. override (set point engage) through pushbutton lights on the front panel.

(5) An Operate-Manual Override Switch. In the manual position the system becomes a position device, manual summed with the valve feedback pot.

(6) A Function Switch on the front panel which allows null balance control and null balance metering of transducer, pre-amp A-1 and D. C. Amplifiers A-2 and A-3.

(7) Meter readings of pressure in %, valve travel in % and load current in milliamps.

A block diagram of the present 57S Configuration is shown in Figure 1.

This unit will have to be modified to meet specific requirements and will consist of adding an appropriate oscillator demodulator module to work with a linear variable differential transformer on the valve. An additional modification on one controller will consist of an emergency override circuit to full stroke the valve.

The output of the 57S is plus/minus 20 MA into a 500 ohm load. The 57S is completely transistorized for high reliability.

The price of the 57S modified is approximately $4,500 each. The estimated price of an appropriate transducer to work with the 57S at cryogenic temperatures is $1,000 each.

Engineering Checkout - It is estimated that approximately two weeks of field engineering would be required. This is highly recommended as problems that generally occur when going from a construction stage to an operational stage can only be handled by factory engineering personnel. This service can be performed at cost of $150.00 per day plus per diem and transportation costs.

Specifications and Supervision of Installation - The specifications for the hydraulic and pneumatic systems would require approximately 100 hours of engineering time at a rate of $15.00 per hour. Supervision of installation can be performed by a Quality Control Engineer familiar with the system and component design concepts and quality control levels required. This work can be performed at a fixed rate of $10.00 per hour.

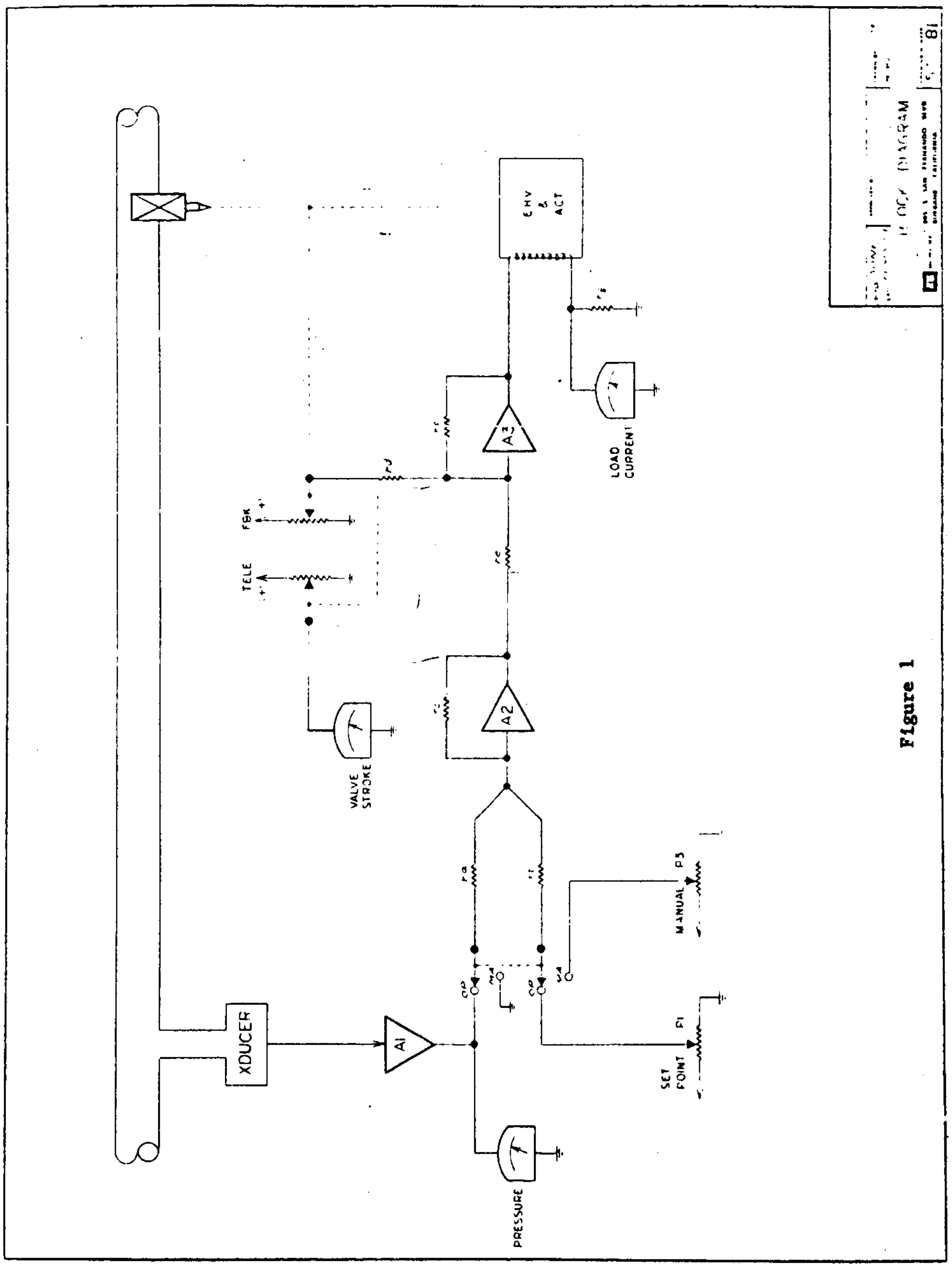

XDUCER
A1
A2
A3
PRESSURE
VALVE STROKE
TELE
FBK
EHV & ACT
LOAD CURRENT
SET POINT
MANUAL
DIAGRAM

Figure 1

WARRANTY: ALL PRODUCTS OF THE COMPANY ARE SOLD AND ALL SERVICES OF THE COMPANY ARE OFFERED SUBJECT TO THE COMPANY'S WARRANTY AND TERMS AND CONDITIONS OF SALE. COPIES OF WHICH WILL BE FURNISHED UPON REQUEST.

FACE-TO-FACE DIMENSIONS IN ACCORDANCE WITH ISA RECOMMENDED PRACTICE RP4.1
ASA STD. FLG. DIM. FOR CAST STEEL, ALLOY, NODULAR IRON AND CAST IRON BODIES
MSS STD. FLG. DIM. FOR BRONZE BODIES

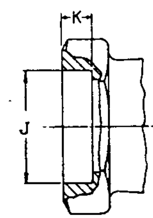
VALVE SIZE (inch)	FLANGED CONNECTIONS: 125 LB.ASA* 150 LB.MSS* 150 LB.ASA**		250 LB.ASA** 300 LB.MSS** 300 LB.ASA**		600 LB.ASA ¼" RAISED FACE		SCREWED OR SOCKET WELD CONN.		SOCKET WELD. END CONN.		C	D				
												D_1	D_2	D_3	D_4: 150 LB. 300 LB. FLGD.	600 LB. FLGD.
	A	A_1	A	A_1	A	A_1	A	A_1	J	K						
½							7	3¼	.855	⅜						
¾							7	3¼	1.065	½						
1	7¼	3⅜	7¾	3⅜	8¼	3⅞	7	3¼	1.330	½	5⅛	6½	9¼		14	14
1¼	8¾	4⅛	9¼	4⅜	9⅞	4 11/16	8	3½	1.675	½	6⅛	7⅜	12¼		14¼	14¼
1½	8¾	4⅛	9¼	4⅜	9⅞	4 5/16	8	3½	1.915	½	6⅛	7⅜	12¼		14¼	14¼
2	10	4⅝	10½	4⅞	11¼	5¼	9¼	4⅛	2.406	⅝	6⅝	7⅝	12⅞		15⅞	15⅞
2½	10⅞	4⅝	11½	5¼	12¼	5⅝	-	-	-	-	8¼	9¼	13⅞		16⅜	16⅞
3	11¾	5¼	12½	5⅝	13¼	6	-	-	-	-	8¼	9¼	13⅞		16⅜	16⅞
4	13⅞	6¾	14½	6½	15½	7	-	-	-	-	8¾①	9¾	15¾		17⅞	18¾
6	17¾	7⅝	18⅝	8⅜	20	9⅝	-	-	-	-	12¾	13⅞	18¼		20½	21¼
8	21⅜	9⅜	22⅝	9⅜	24	10⅝	-	-	-	-	13⅞	15	20⅞		22⅞	24
10																
12																

* FLAT FACE
** 1/16" RAISED FACE
D_1 – PLAIN BONNET
D_2 – RADIATION FIN BONNET
D_3 – BELLOWS BONNET
D_4 – EXTENSION NECK BONNET

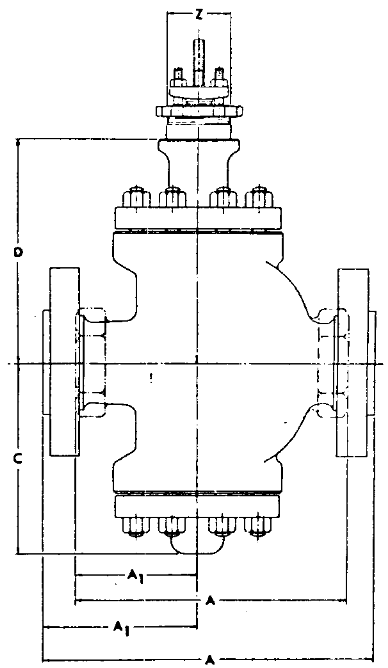
① 8¾" FOR 150 LB. AND 300 LB. FLANGED VALVES. 9⅝" FOR 600 LB. FLANGED VALVES.

BODY DATA AND ACTUATOR DRAWING REFERENCE CODE

VALVE SIZE (inch)	VALVE TRAVEL	STEM CONN. SIZE	ACTUATOR DRAWING REFERENCE: BONNET MOUNT DIAM. Z	DIAPHRAGM TYPES		ON-OFF PISTON		POSITIONING PISTON		MANUAL
				STANDARD	OVERSIZE	STANDARD	OVERSIZE	STANDARD	OVERSIZE	STANDARD
½, ¾, 1	¾	½	2⅛	b	c	b	c	b	c	a
1¼, 1½, 2	1⅛	½	2⅛	c	d	b	c	b	c	a
2½, 3, 4	1½	⅝	2¾	b	c	a	b	a	b	a
6, 8	2¼	⅝	3¾	b	e	c	d	a	b	b
10, 12	3½	⅞	3¾	c	d	e	f	c	d	c



☐ SOCKET WELD. END CONN.



☐ FLANGED CONNECTIONS
☐ SCREWED CONNECTIONS

THIS PRINT CERTIFIED CORRECT

CUSTOMER ______________________ P.O.No. __________
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Water Brake Power Absorber

An unusually large water brake (Absorption Dynamometer) has been designed and built by Kahn & Co., of Wethersfield, Connecticut. It is nine feet long by six feet wide by six feet high and weighs 6000 pounds. The brake can absorb approximately 30,000 hp at 6000 rpm, which makes it well suited, when two units are used, for absorbing the power output of the turboexpanders considered for this study program. It can be used at speeds up to 8000 rpm, will operate in either direction of rotation, and has a controllable range-ability of approximately 15 to 1 with a maximum torque capacity of 30,000 ft-lb. The maximum conditions of both horsepower and speed are shown on Figure 24. Apart from this point, there is no other breakdown condition.

Water flow requirements for this unit, or any other water brake dynamometer, may be calculated by using the value of 4 gallons per hour per horsepower. On the assumption that each water brake will be required to absorb 28,000 hp maximum, one unit will require a water flow of 1900 gpm, and the second unit from 0 to 1900 gpm, depending on the total horsepower required. The maximum allowable case pressure is 350 psi. The casing will be protected by a relief valve in the event that this pressure is exceeded. It is recommended that the water temperature at the outlet of the brake does not exceed 180°F; and as the flow rate of 4 gallons per hour per horsepower is based on a 70°F rise through the brake, it follows that the water flow can be substantially reduced, if the inlet water temperature is low.

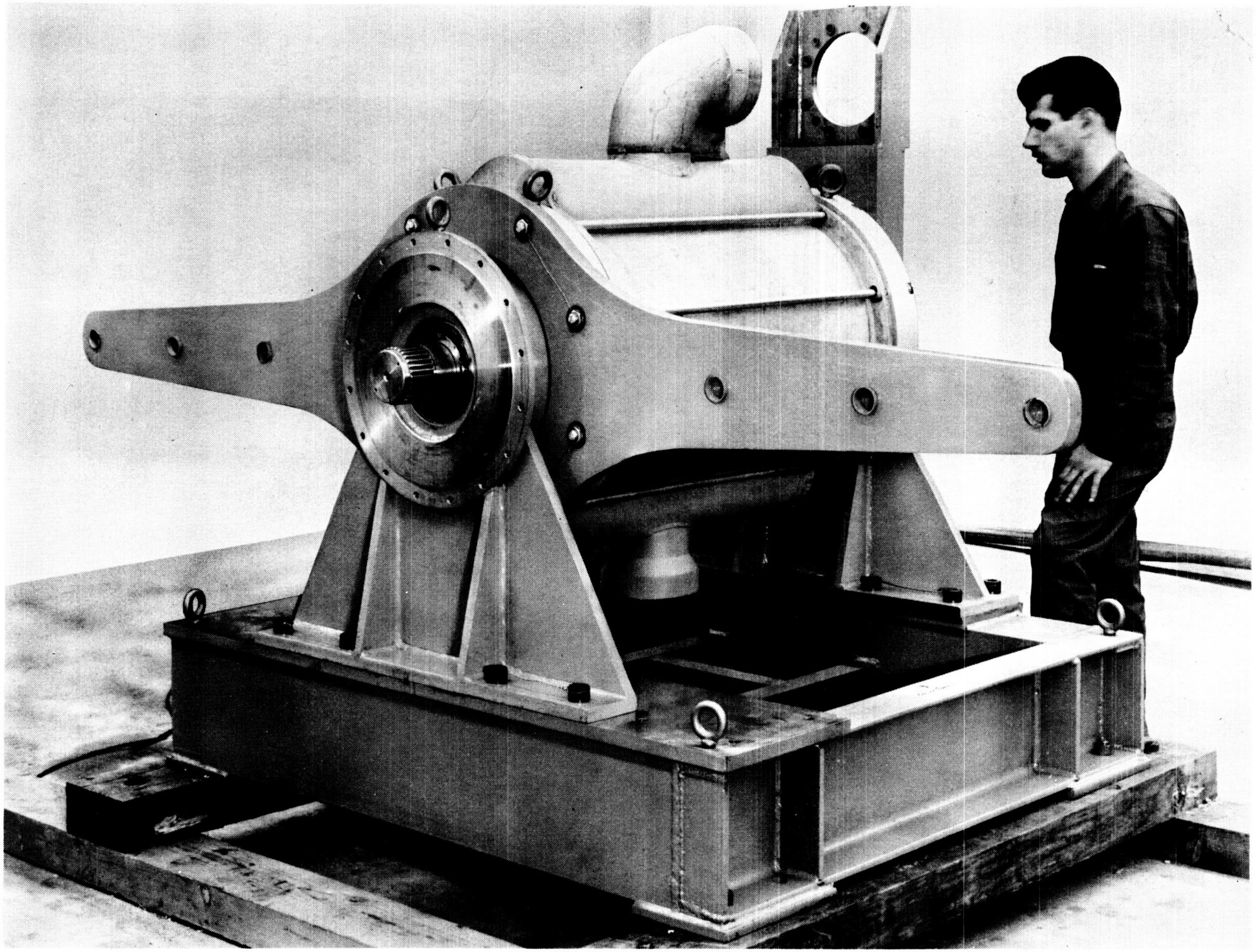
The moment of inertia of each rotating assembly is 2.396 lb-ft per sec^2.

Each water brake will be supplied with two remotely controlled flow control valves, one on the inlet and one on the outlet of the water brake. The control of these valves will be graduated such that once the information is acquired as to their setting position for the specified parameters, then controls may be preset to these empirically obtained positions. For a particular speed-horsepower relationship, the brake performance is affected only by a change of inlet water pressure, assuming that the control valve positions are not changed. In order to minimize the effect of pressure fluctuations in the water supply header, a pressure regulator will be installed upstream of the inlet flow control valve, which will maintain a steady inlet pressure to the flow control valve of 30 psig.

The control system will be overridden by a pressure relief valve, in the event the case pressure exceeds its design maximum. It is assumed that the turbine will be governed such that the water brake will not exceed 8000 rpm. It should be noted that the unit that is in operation at General Electric was subjected to a spin test of 9000 rpm before being placed in service.

Each dynamometer will be mounted on a frame which should be rigidly grouted into a concrete pad with a trench for the water outlet. Depending on the length of run required for the six-inch water lines, both for inlet and outlet, it is very roughly estimated that the installation costs should not exceed $5000. The cost of two Water Brake Dynamometers, Kahn Model 061-040M, with rigid base mounting and frame would be $110,000.

Regarding the disadvantages of the water brake dynamometer as compared to jet engine compressors or centrifugal air compressors, we are firmly of the opinion that following the experience gained in the existing installation of this water brake, the control system should present no problem of sensitivity in comparison to other types of loading devices. Also, as long as there is a source of water pressure of 40 psi or more available, minor fluctuations will be nullified by the pressure regulator.



KC-061-040 WATER BRAKE DYNAMOMETER
Manufactured by Kahn & Co., Inc., Hartford 1, Conn.

Maintenance

The total test time, as indicated by the problem statement is only 35 hours, and the number of operating cycles is 440 for a five year period. In comparison with the operation of conventional rotating machinery, this length of time and number of cycles is very low, and no special maintenance of the turbine system and controls will be required. This conclusion is valid provided that the operating procedures for the M-1 pump include the provision and maintenance of a clean liquid hydrogen circuit so that particles which may be harmful to the bearings and seals will be eliminated.

In view of the untested status of the turbine, the presence of a highly qualified turbomachinery engineer during the entire period is recommended. The duties of this position would be to control the operation of the turbine system and to monitor the operational condition of the various components from the data supplied by the instrumentation, such as bearing temperatures, leakage detectors, vibration sensors and the usual pressure-temperature inputs. This cost is estimated at $30,000 per year or $150,000 for five years.

As discussed previously, the maintenance cooldown costs will be negligible due to the essentially "free" cooling obtained from the use of storage tank boil off vapors. Even without vapor cooling, the vacuum jacketing will reduce the cooldown loss to less than $5500 of LH_2 per year. Therefore this item was considered negligible.

For the heat exchanger system, the maintenance requirements would be negligible. The single control valve could readily be operated by the M-1 pump test crew in accord with a relatively simple procedure during start up or changes in pump test point settings.

Amortization

On the basis of the original problem statement of 7 tests per month, it appears that the turbine-compressor system will save its costs in less than 4 months of operation. However, the test schedule, as given in Section II, is not uniform, and the build up to an average or maximum usage will require nearly two and one half years. On this basis the recovery of costs will require about one and one half years from start up.

The timing for the heat exchanger system will be nearly the same, since this system has roughly onehalf the potential savings at one half the cost.

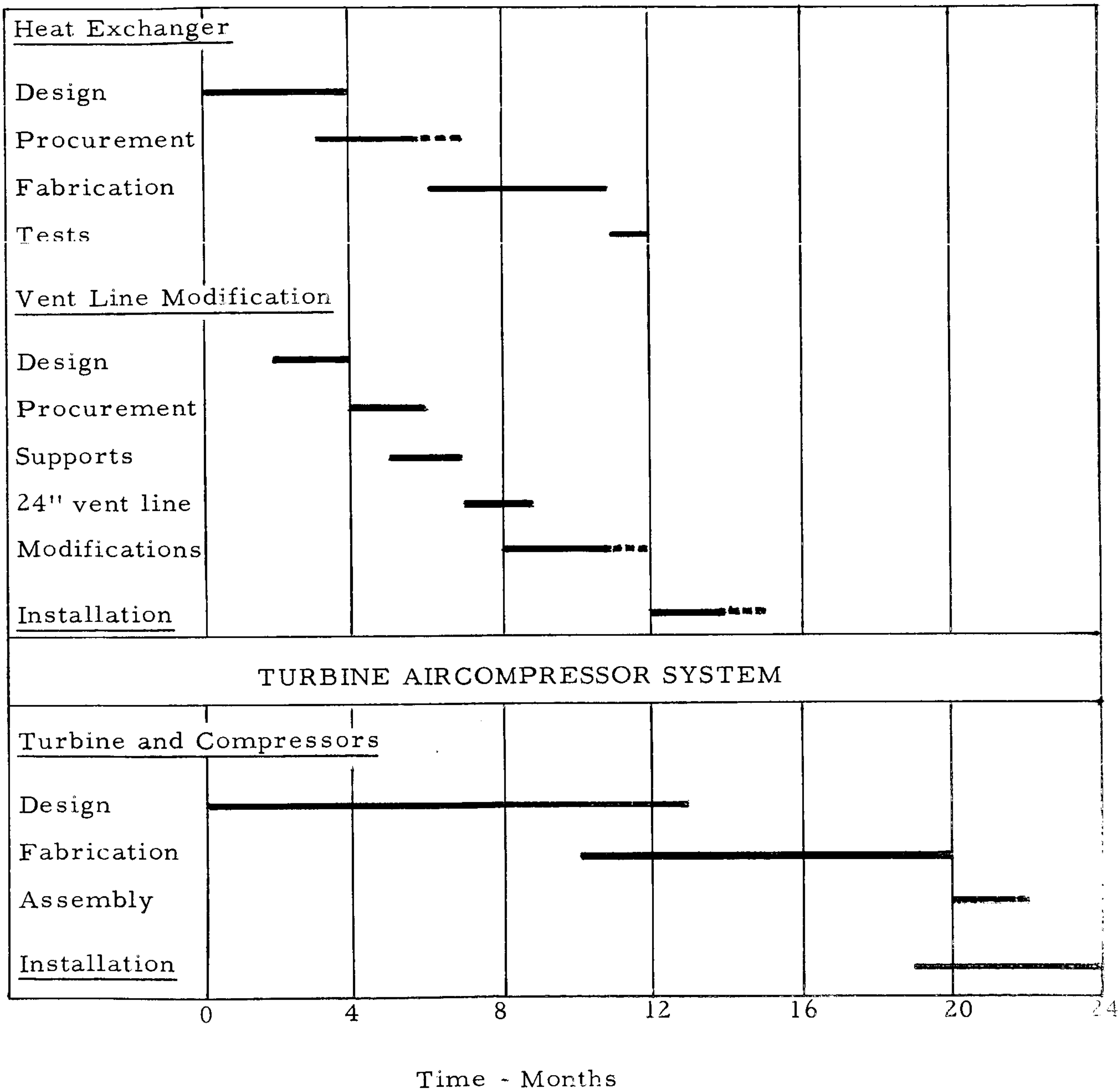
The following table gives the time required for the recovery of costs through the savings in liquid hydrogen. The first column is based on a frequency of 4 tests per month and the second column is based upon the use rate given in figure 14 of section II. Assuming a 6% rate of interest charge during this test period, the total costs are increased to the amounts shown.

Frequency of Tests		
Turbine System	4 Tests per month	Figure 14 (Section II)
Time for saving costs in LH_2	5 months	20 months
Total cost at 6% interest	1,670,000	1,725,000
Heat Recharger System		
Time for saving costs in LH_2	6 months	24 months
Total cost at 6% interest	862,000	900,000

IX SCHEDULES

The following bar charts present tentative schedules for the heat exchanger and the turbine-Air Compressor systems.

HEAT EXCHANGER SYSTEM



Down Time

The heat exchanger system will require a relatively long down time for modification of the present vent system. The timing would be governed by the status of the heat exchanger fabrication and would not be started until the entire procedure of modification and installation could be accomplished as a continuous effort. The total time is estimated to be from 3 to 5 months.

For the turbine-air compressor system all of the main piping can be installed prior to actual tie-in so that a minimum of down time is anticipated. Conceivably the tie-in can be accomplished in less than one month.

Obviously, the down time may be greatly dependent upon the experience and capabilities of the contractor and subcontractors engaged for this effort. From the overall economic standpoint it may be preferable to consider this capability, rather than the conventional "low bid" as a basis for contract award.

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Los Alamos Scientific Lab (1)	Los Alamos Scientific Laboratory Los Alamos, New Mexico Attention: Dr. F. Edeskuty
USAF Rocket Propulsion Lab (1)	USAF Rocket Propulsion Laboratory RPRRE Edwards Air Force Base Edwards, California Attention: H. V. Main
Aerojet-General Corporation (2)	Aerojet-General Corporation Liquid Rocket Plant P. O. Box 1947 Sacramento, California Attention: J. Laws D. Price
C. F. Braun Company	C. F. Braun Company 1000 No. Freemont Avenue Alhambra, California Attention: William Jack
Paul Hardeman, Inc.	Paul Hardeman, Inc. 10579 Dale Avenue Stanton, California Attention: M. S. Winders
Stearns-Rogers Corporation	Stearns-Rogers Corporation 600 No. Broadway Attention: Mr. William Boyer Mr. William Lang

Linde Company	Linde Company 2770 Leonis Blvd. Los Angeles 58, California Attention: Mr. P. J. Murto
O. Kahn and Company	O. Kahn and Company P. O. Box 516 Hartford, Connecticut Attention: Mr. Irving Kahn

ERRATA SHEET

FINAL REPORT
LIQUID HYDROGEN RECOVERY SYSTEMS STUDY
NASA CR-54010

Page	Line	Correction
2	17	$1,700,000
9		Figure 2
		Add: h_f at left end point of line h_{fg}
		Use: h_f' in place of h_f, at end point of line h_{fg}'
16	27	Delete: . . . , and this optimum does exist , . . .
18	21	Figure 4
19	8	Figure 4
20	15	100 psi $1,300,000
	16	$9,000,000
	17	$20,000,000
28	19	---steady-state---
38	20	New paragraph heading: Cost of Aerojet System
63	Title	KAHN WATER BRAKE (2 UNITS)
80	--	Delete: at center of drawing, scale, 1/8 in. = 1 ft
114	19	Spelling: sequence
118	5-10	Use h_f in place of h_o
123	11	$\dot{w}$ should be $\dot{w}_\beta$
124	14	Integration limits should be, $\int_{T_e}^{T_a}$ $\overline{}$ $\int_{T_o}^{T_e}$
124	Bottom	Add: $T_e = T_o + \frac{\Delta T_v}{2}$, approximately
129		Add: $\left.\begin{array}{l} T_{2p} \\ T_{2f} \\ T_8 \end{array}\right\}$ outlet temperatures